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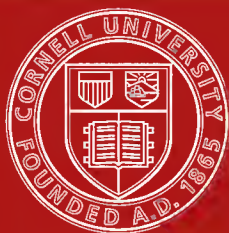
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BY CHARLES L. HUBBARD, B.S., M.E.

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McGRAW-HILL BOOK CO., Inc.

239 WEST 39TH STREET

NEW YORK CITY

POWER, HEATING AND VENTILATION

PART III

COMBINED POWER
AND HEATING PLANTS

A Treatise for Designing and Construct-
ing Engineers, Architects and Students

BY

CHARLES L. HUBBARD, B.S., M.E.

Consulting Engineer

FIRST EDITION

McGRAW-HILL BOOK COMPANY, INC.

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PREFACE

The subjects covered in the following chapters relate especially to the power and heating equipment for buildings of large size, such as office buildings, department stores, municipal buildings, hotels, hospitals and institutions, theatres, technical and trade schools, loft buildings, railroad stations, machine shops, foundries, paper and textile mills.

The mechanical equipment in buildings of this kind is more or less complex, depending upon their size and use, and may provide for the generation of power, lighting, elevator service, refrigeration, pumping, fire protection, air compression, vacuum cleaning, drying, water heating, cooking, sterilizing, etc., not to mention the special machinery used in various manufacturing processes.

In some cases only a few of these requirements may be necessary, while in others, like the modern office building or hotel, practically all of those noted are called for.

It has been the aim in the present volume to treat the various problems somewhat broadly and with special reference to the steam requirements. Equipment relating to the generation and use of steam has been considered in some detail, while other matters, such as lighting, elevator service, refrigeration, etc., have only been touched upon as they relate to the steam side of the problem. It has been assumed that the reader is already familiar with the usual details of heating and ventilating design, hence, attention has only been given to the special requirements of the buildings under consideration. In the later chapters a study has been made of the relation between power and heating, and methods given for determining the results to be expected by combining the two in a single plant under varying conditions.

The object sought has been to enable one to take a comprehensive view of problems of this kind, to decide upon the best system of heating to be employed, to determine what advantage, if any, may be gained by generating power upon the premises, and

also what type and arrangement of power equipment will be best fitted to meet the special requirements.

Furthermore, the various steps in the design of a plant of this kind have been taken up in logical order and the computations worked out in detail to show their application to the various conditions which may arise in practice.

In a general way this volume supplements those which precede it in the same series, and will be found to contain a large amount of practical data relating to the actual design of combined power and heating plants not usually included in books which treat these subjects independently.

Special thanks are due to the editors of various magazines for permission to use material which has appeared over the author's name in their publications. Among these are included the following, together with a list of the chapters from which portions have been published.

American Architect, Chapter 1.

Brickbuilder, Chapter 11.

Engineering Magazine, Chapter 3.

Isolated Plant, Chapters 5, 10, 12.

Metal Worker, Chapter 4.

National Engineer, Chapters 2, 3, 6, 7.

Power, Chapters 2, 6, 7, 8.

Practical Engineer, Chapter 8.

Credit is also due the Warren Webster Co., for data on humidity and air cooling.

Of the 220 cuts, 130 are from original drawings, prepared especially for this work by the author. The others have been redrawn from cuts taken from magazines and catalogues, as these seemed to fulfill the specific purpose sought, better than original drawings.

A list of cuts taken from various magazines is given below:

Engineer, Figs. 100, 101, 132, 133, 160, 165, 198.

Engineering Record, Fig. 30.

Machinery, Figs. 7, 99, 109, 127, 129, 130, 176, 177, 178, 179, 180, 182, 183.

Metal Worker, Figs. 52, 54.

Plumber's Trade Journal, Figs. 143, 144, 145.

Power, Fig. 199.

Practical Engineer, Figs. 153, 154, 181.

Steam, Figs. 189, 190, 191, 192, 193.

A list of cuts redrawn from trade catalogues is given in the

back of the book for identifying the make of the different pieces of apparatus described.

Thanks are extended to the publishers of the various magazines noted, and to the manufacturers of the equipment shown, for the use of cuts as mentioned above.

Data and tables appearing in forms similar to the originals from which they were taken, are credited in the text to the source of information.

It has been the intention of the author to give credit in every case where due, but if through oversight this has not been done, it will be considered a favor to have such oversight reported in order that it may be corrected in future editions.

CHARLES L. HUBBARD.

Boston, Mass.

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PART III

COMBINED POWER AND HEATING PLANTS

COMBINED POWER AND HEATING PLANTS

CHAPTER I.

HEATING AND VENTILATING REQUIREMENTS FOR DIFFERENT TYPES OF BUILDINGS.

The following chapter discusses briefly some of the principal requirements to be considered in designing heating and ventilating plants for different types of buildings. In connection with this discussion, various methods are given for meeting these requirements in a manner to conform with the standards of present engineering practice.

Office Buildings.

1. General Arrangement.—The modern office building usually consists of three sections, the basement or sub-basement containing the mechanical plant; the street floor, devoted to banking rooms with suites of private offices above; and the remainder of the building made up of small rooms arranged in a similar manner upon each floor.

2. Methods of Heating.—The upper section is commonly warmed by direct radiation, it not being practicable to combine heating and ventilation in buildings of this kind owing to the large number of rooms and the necessity of individual temperature regulation in each. The banking rooms and private offices are generally provided with ventilation and the heating is usually done by means of hot air, supplemented by direct radiators, where needed. When a ventilating fan supplies more than one room

it is customary to raise the temperature of the entering air to 72 or 74 degrees at the main heater and provide the additional heat either by direct radiators or by secondary stacks placed at the bases of the various flues, thus giving to each room an independent means of temperature control. In buildings of small size, indirect gravity heaters are often used for warming the banking rooms, although not so effective as a fan system.

3. Location of Radiators.—In the smaller rooms of the upper section the radiators are generally placed beneath the windows for the following reasons: First, it is the coldest part of the room and this arrangement serves to prevent down-drafts caused by the large glass exposure and the inleakage of air around the sashes; second, the partitions are often changed to suit the tenants, and with the radiators in this location there will be no interference, whatever the arrangement of the rooms; third, it simplifies the running of supply and return risers, making the layout practically the same on each floor. There may be instances, however, where it is desirable to keep the windows free, in which case the radiators may be set against the side walls, or partitions, without interfering with their effectiveness to any great extent.

In steam-heated rooms, having two or more windows, it is best to divide the radiating surface, placing a unit beneath each window. This arrangement makes it possible to proportion the heating capacity to the outside temperature by shutting off one or more of the radiators in mild weather. In small rooms, where a single radiator is used, the same result may be secured by dividing it into two separately valved sections and connecting with the riser, as shown in Fig. 1. This, in effect, gives two independent radiators while having the appearance of only one. It can be employed, however, only with a single-pipe connection, but as this method is widely used in low-pressure gravity heating it is available in a large number of cases. Vacuum and hot-water systems, requiring the two-pipe connection, have other means of regulation, so the divided radiator is not necessary under these conditions.

Entrances and corridors may be warmed either by direct or indirect radiation. A good arrangement for this purpose is to conceal the heaters by grille work and force warm air through

them by connecting with the fan system. This serves to supply fresh air to the corridors and also reduces the radiating surface by increasing its efficiency. The larger part, if not all, of the corridor radiation may be placed on the first floor. These rooms have but little, if any, outside exposure and the stairways and elevator shafts act as flues for carrying the heat to the upper stories. In special cases, where there is considerable outside glass exposure, it may be necessary to supplement this with a small amount of direct radiation suitably distributed on the various floors. In general, however, the first arrangement will provide

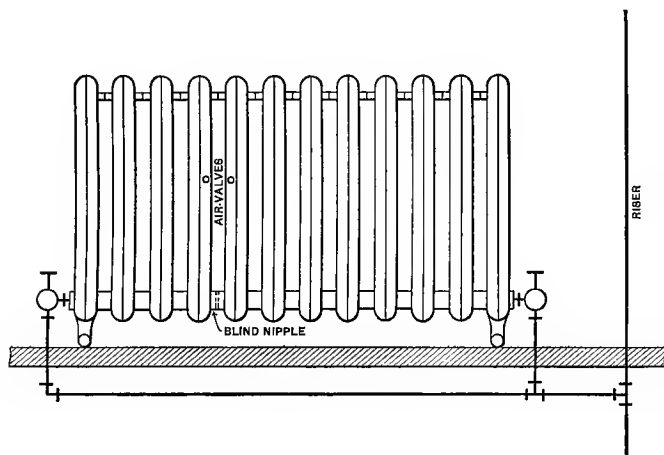


Fig. 1. Divided Radiator for Hand Regulation.

ample heat and is more simple to install. When a fan is used in connection with this work provision should be made for gravity rotation through the heaters at night when the power is shut off. Miscellaneous rooms, such as lunch rooms, barber shops, toilet rooms, basement storage rooms, etc., are usually heated by direct radiation located according to the positions of the risers and the general arrangement of the fixtures.

Skylights may be warmed by carrying coils of pipe around the enclosed chamber between the two sashes.

4. *Radiator Shields.*—Cold down-drafts in front of high windows in banking rooms and similar locations are often difficult to

overcome with ordinary direct radiators. In cases of this kind good results may be obtained by the use of plate-glass shields as shown in Fig. 2. These form a flue which delivers a current of warm air at a sufficient height to break up or deflect the downward movement of cool air before it reaches those sitting below.

Other methods of overcoming down-drafts are considered under ventilation.

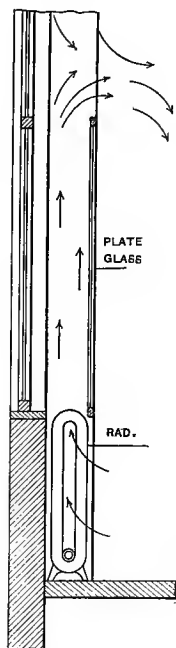


Fig. 2. Plate Glass Radiator Shield.

5. Types of Radiation.—The various forms of cast-iron sectional radiation are used almost exclusively in this class of work. Direct radiators should be of plain pattern and chosen with reference to the positions they are to occupy. A certain relation should be maintained between the height and length in order to give a symmetrical appearance, and two-column radiators are somewhat more efficient, for a given surface, than deeper ones. Where the space below the windows is limited, low radiators of special design are often required, a typical form for this purpose being shown in Fig. 3.

Modern fire-proof building construction usually makes it necessary to carry the run-outs to the radiators above the floor. When they are located at some distance from the risers it will be advisable, in gravity systems, to use radiators having extra high legs in order to secure sufficient pitch for proper drainage.

When a vacuum system is employed this precaution is not necessary, as the suction in the return will keep the pipes clear of condensation. Wall radiators are especially adapted to the warming of toilet rooms and lavatories because they leave the floors free for easy cleaning or flushing.

Stacks for indirect gravity heating, and supplementary heaters for use with fan systems, are best made up of pin radiation or sections of similar form.

6. Room Temperature.—All offices and banking rooms should be warmed to a uniform temperature of 70 degrees in the coldest

weather. Although a corridor temperature of 60 degrees, or less, is ample for the comfort of those entering or leaving a building, it is best to maintain a temperature approximating that of the offices on account of those passing through to other rooms, toilets, etc., without extra clothing. Toilet rooms containing lavatories or bathing facilities should be warmed to 70 degrees, while in those of a semi-public nature, used by people dressed for the street, a temperature 55 to 60 degrees is sufficient. Private toilets usually have little, if any, outside exposure and are not commonly provided with heat.

7. *Minimum Outside Temperature.*—The minimum outside temperature upon which the capacity of a plant should be based will depend upon the geographical location of the building. The United States Government, in the design of its Federal buildings for northern cities, bases its calculations for heating capacity

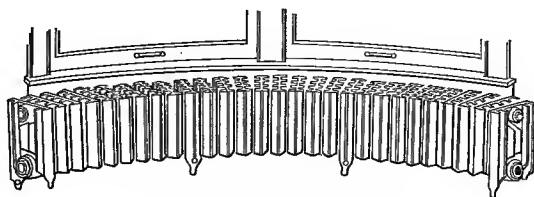


Fig. 3. Special Window Radiator.

upon the lowest recorded temperature in the past ten years, while for southern cities, the lowest recorded temperature plus 10 degrees is taken. Although it is very important that an office building be properly warmed at all times, it hardly seems necessary to provide a plant of sufficient capacity for the exceptional low records which occur only once in several years. In New England and the Middle States a minimum temperature ranging from -10 degrees to 0 is commonly assumed. When it is desired to provide against possible low records, the ducts may be so arranged as to allow of the air being rotated within the building, thus sacrificing ventilation for additional heat, temporarily. Dampers or doors should be provided so that by closing off the outside supply, air may be drawn downward through corridors and stairways, passed through the main heater, and delivered again into the lower corridor at a comparatively high temperature.

This is much more economical, both in cost of installation and operation, than to increase the direct radiation throughout the building. When the building is once thoroughly warmed, a well-planned system should be able to maintain a comfortable temperature, even with the outside conditions considerably more severe than those for which the plant was originally designed.

8. *Systems of Heating.*—Both steam and hot water are employed in this class of work. When the former is used it may be circulated under a low pressure, returning the condensation to the receiving tank by gravity, or a vacuum system may be employed. In combined power and heating plants using the former arrangement the pipes should be of such size that the steam will circulate freely through all parts of the radiating system with a pressure not exceeding 2 pounds, in order that the back-pressure on the engines shall not be materially increased over that when exhausting to atmosphere. Gravity circulation is adapted to buildings of small and medium size, requires no special devices in the way of return valves, vacuum pumps, or exhausters, and also permits of the single-pipe connection.

In larger buildings the circulation is apt to be sluggish, especially when steam is first turned on, and higher pressures are required unless the pipe sizes are considerably increased. In plants of this kind, using exhaust steam, it is usually best to employ a return-line vacuum system, which accelerates the circulation without increasing the pressure, does away with air valves, and makes possible a certain amount of temperature regulation by using graduated valves at the supply end of the radiators.

Hot-water heating, under forced circulation, is adapted to the same class of buildings as the vacuum steam system. The exhaust is utilized in a tubular heater, without back pressure, and the water is forced through pipes of moderate size by means of a circulating pump. In the "vacuo" system, so called, the heater is made to take the place of a condenser, either wholly or partially, thus adding to the power or economy of the engines. By circulating the water at different temperatures, according to outside conditions, a certain amount of general temperature control is possible which is under the direct charge of the engineer, thus adding materially to the economy of operation.

9. *Temperature Control.*—In the best class of work, and whenever funds are available, the heating system should be equipped with automatic temperature regulation. This is important, not only as regards the comfort and health of the occupants, but also reduces the cost of fuel, unless there is a surplus of exhaust steam available for all heating purposes.

The methods of temperature control previously mentioned are of considerable advantage in a limited way, but are only make-shifts when compared with an efficient system of pneumatic regulation. Devices dependent upon the attention of the occupants are likely to be neglected, and temperature control is usually at-

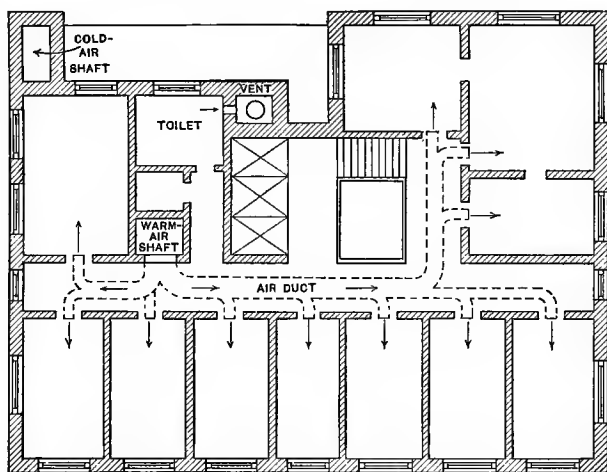


Fig. 4. System of Ventilation for Office Building.

tempted by opening and closing the windows, resulting in disagreeable and often dangerous drafts and waste of heat.

Whatever the system of heating and ventilation, it is a most excellent plan to open the windows for a short period occasionally and flush out the rooms with fresh air, but it is not a good idea to sit in a draft nor to remain in an over-heated room for any length of time.

10. *Piping Arrangement.*—When the lower floors are used for banking rooms or similar purposes it is desirable to conceal all steam and return piping leading to the upper part of the building.

This can best be done by supplying the first floor radiators from a separate up-feed system, and the remainder of the building from an independent line, on the down-feed plan, as illustrated in the following chapter.

11. *Ventilation.*—It has already been stated that ventilation is not usually provided for all floors in buildings of this type. There are instances, however, where the owners desire to have the entire building ventilated, or at least a considerable portion of it. When this is done the arrangement shown in Fig. 4 is usually employed. This consists of a main flue, centrally located, and extending to the top of the uppermost floor to be ventilated. Horizontal ducts connect with this vertical shaft at each ceiling, of sufficient size to supply the entire story. These ducts are con-

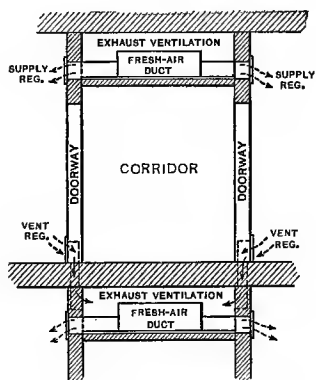


Fig. 5. Detail Showing Ducts over Corridor.

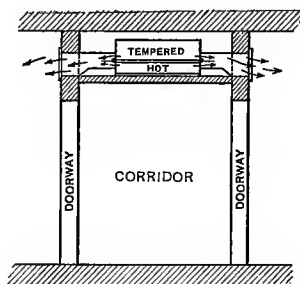


Fig. 6. Detail of Double-Duct System.

cealed by a false ceiling or furring at the upper part of the corridor and deliver fresh air to the rooms through registers usually placed over the doors. Sometimes the furred-in space around the duct is utilized for exhaust ventilation being connected with a fan located upon the roof. This general arrangement is shown in Fig. 5. With certain forms of building construction it may be necessary to place automatic fire dampers over the openings leading from the main supply and vent shafts at each story. This, however, is a detail to be decided by the fire underwriters for the district in which the building is to be erected. The usual

plan, however, is to simply supply fresh air to the smaller rooms and depend upon leakage for exhaust ventilation.

Heat is commonly furnished by direct radiation independently of the ventilating system which supplies air at 2 to 4 degrees above the normal temperature of the room.

Sometimes the double-duct arrangement is employed for limited sections of a building where the rooms are elaborately finished and it is desired to eliminate all direct radiation. The hot air, in this case, is carried below the tempered air as shown in Fig. 6, and the two mixed in the right proportion by a graduated damper thermostatically operated.

12. Ventilation of Banking Rooms.—The best results are obtained in rooms of this kind by the use of fans, both for supply and exhaust. The location of these will depend somewhat upon the arrangement of the building. Ordinarily the supply outfit is placed in the basement, although in some cases it may be desirable to locate it above the rooms and discharge the air into flues leading downward. Vent flues from the first floor are usually gathered at the basement ceiling and connected with an exhauster discharging into a special shaft leading to the top of the building. There are various ways of admitting the warm air to the main banking room. If the system is to be used for heating only, and not for cooling in the summer time, it is a good plan to bring in a considerable proportion of the warm air through long narrow slots in the window sills, and through elevated registers in or near the outer wall. A certain amount should also be supplied to the public space by means of grilles along the inner wall or through centrally located columns.

Exhaust ventilation should be through grilles or registers placed near the floor, part in the base of the counter and part in both outer and inner walls, if the room is of considerable width. In long narrow rooms, the fresh air is often brought in through a series of registers, about 8 feet from the floor, located in the outer wall and the exhaust taken off through openings near the floor along the opposite side of the room. An examination of the plans of a considerable number of installations of this kind, representing the most recent practice, shows a considerable variation in treatment as to the location of air inlets and outlets. Much seems to depend upon the available space for flues, and the only

general rule followed appears to be the admission of air at an elevation of 7 or 8 feet and its removal at, or near floor. In several of the plans examined the larger proportion of both supply and vent registers were located along the same wall. When a washer or other cooling device is employed, and the air is admitted in summer at a temperature considerably below that of the room, a somewhat different flue arrangement must be provided from that noted above. Ordinarily the space along the outer walls, next the windows, is occupied by desks, and if the cooler air were admitted in the usual manner it would at once fall upon the heads of those sitting below.

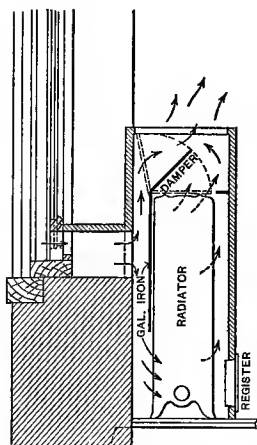


Fig. 7. Encased Radiator for Window Ventilation.

With systems of this kind the air should be introduced along the inner walls, or at the opposite side of the public space, and exhausted at the outer walls, after having become diffused and raised to the normal temperature of the room. This arrangement works equally well with warm air, provided direct radiators and shields are placed in the windows as shown in Fig. 2. Whatever the method of ventilation, a sufficient number of direct radiators, or rotation heaters, should be provided to warm the room to a comfortable temperature when the fans are not running. In general, we may say

that the air should be heated to about 72 or 74 degrees at the fan and delivered to the room partly at this temperature and partly at a higher temperature by passing through re-heaters located beneath the window flues, these being available as rotation heaters, when the fan is not running, by the manipulation of switch dampers.

Private offices may be heated by direct radiation either encased or exposed, or by re-heaters at the bases of the supply flues. Sometimes direct radiators are screened and the air supply brought in back of them.

An arrangement frequently used for the ventilation of isolated offices is shown in Fig. 7, and consists of a direct radiator en-

cased so as to take air from beneath the raised sash of a window. A mixing damper is provided for regulating the temperature, and a rotation damper for night use or at times when ventilation is not required.

13. Volume of Air Supplied.—When the probable number of occupants is known it is best to proportion the air supply upon this basis, allowing at least 40 cubic feet per minute each.

When this data is not available a certain number of changes per hour may be provided. Under ordinary conditions from four to five changes should be furnished in the smaller rooms of an office building and from three to four in banking rooms.

If the rooms are very high the number of changes may be reduced.

In a building recently erected, a banking room 58 feet in height

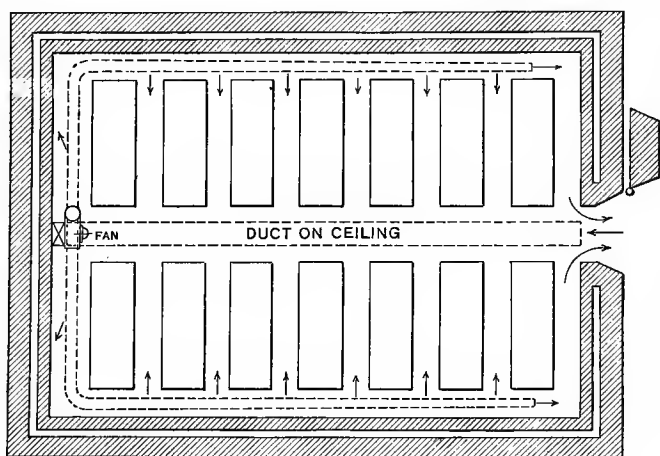


Fig. 8. Ventilation of Safety Deposit Vault.

was satisfactorily ventilated with two changes per hour. When designing a ventilating equipment for rooms of this type the capacity of the exhaust system may be reduced to about 0.6 to 0.7 that of the supply system, as a considerable amount of air will find its way out by leakage.

14. Vault Ventilation.—Safety deposit and storage vaults require special treatment, because, when closed at night, there must be no access through air ducts.

There are various ways of overcoming this difficulty, one of which is shown in Fig. 8. In this case a small centrifugal fan is supported upon the wall at the rear of the vault and arranged by means of a shallow ceiling duct to draw air from the upper part of the doorway. This air being taken from a well-ventilated corridor, and therefore of good quality, is delivered at several points in the rear and along the sides of the vault and passes out under a slight pressure through the lower part of the doorway. Another plan is to omit the special fan and connect with the regular ventilating system by means of a removable sleeve which is disconnected at night when the doors are closed.

15. Ventilation of Special Rooms.—Exhaust ventilation is provided for toilets, kitchens, lunch rooms, etc., by means of separate fans in order to make them independent of the banking rooms. Whenever a room is likely to contain smoke or unpleasant odors the flow of air should be outward, hence, a supply is usually drawn from the corridors by suction and discharged outboard instead of being furnished under pressure by a special supply fan.

Rooms of the above type require from six to eight changes per hour under average conditions and often more in special cases.

16. Air Washing and Cooling.—Air washers are now generally included in the ventilating equipment of all buildings of this type, both on account of the more healthful quality of the air supply and for protecting the building and its contents against injury from soot and dust. These devices are easily equipped for humidity control and may be used for air-cooling in the summer.

The amount of cooling effect will depend somewhat upon local conditions and may be increased by taking the spray water from the city mains or deep wells. In some cases a refrigerating plant is installed and brine or other cooling medium circulated through a series of coils over which the air is drawn by the fan.

Under ordinary conditions the temperature of a well-ventilated room may easily be kept 10 degrees lower than the outside air.

Department Stores.

17. General Method of Heating.—Department stores are usually heated by a combination of direct and indirect radiation, the latter being installed in connection with the ventilating fans.

The most crowded departments are generally those on the first floor and in the basement, and mechanical ventilation is usually confined to these sections of the building. While there may be special cases where a generous air supply is required on some of the upper floors, it will be found, as a rule, that ample ventilation will be secured by inleakage around windows, supplemented by the general upward movement of the air within the building.

18. Method of Air Supply.—The entire floors in buildings of this kind are usually unobstructed by partitions, except those extending only part way to the ceiling, and must therefore be treated as single rooms of large size. Here, as in the case of banking rooms, the flue arrangement will depend somewhat upon the building construction and the available space. When the windows are placed 7 or 8 feet above the floor, as is often done to make room for cases and shelving, the warm air may be brought in directly beneath the sills thus offsetting the effect of down-drafts from the exposed glass area.

Another arrangement, often employed with good results, is to bring the flues up on the first row of columns and discharge the warm air toward the outer walls, as shown in Fig. 9. A portion may also be brought in through registers located in the bases of the counters, preferably at the ends where people are not likely to stand.

When the building occupies a corner, or has more than two outside walls, special provision should be made to distribute the heat over the entire exposure as evenly as possible. In many instances the building is surrounded by others and has only a front and rear exposure, in which case the warm air is delivered near the outer walls with the larger portion at the front, adjacent to the main entrance.

Air entering through registers at an elevation of 7 or 8 feet may have a velocity of 500 to 600 feet per minute, but in counter and base-board inlets it should be limited to about 300 feet.

The method of supplying air to the basement will depend somewhat upon the purpose for which it is used. If the appearance is not important, distributing ducts of galvanized iron may be carried on the ceiling, with inlets delivering air at frequent intervals near the outer walls. When overhead ducts are undesirable, the air

may be brought in through wall flues in the form of pilasters supplied from ducts carried at the ceiling of the sub-basement.

19. *Fans and Heaters.*—These are generally located in the basement or sub-basement, and the cold air brought down through a special shaft from a sufficient height to avoid drawing in an excess of street dust. Separate units are usually employed for each floor, and the entire radiating surface is sometimes

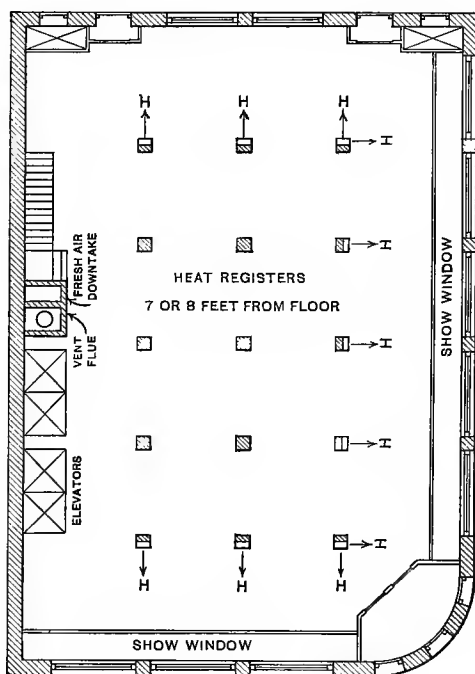


Fig. 9. Ventilation of a Department Store.

massed in the main heaters. In very large buildings a pair of fans may be used, one supplying each half of the floor, and each heater being regulated independently by a thermostat placed in that part of the room which it supplies. With this general arrangement night warming must be provided for by air rotation through the heaters, taking the supply from the basement or other interior source. If a sufficient air circulation is not obtained by

gravity it will be necessary to operate the fans at a very low speed during cold nights.

In order to secure an effective air movement by gravity rotation it will be necessary to place the main heater directly at the base of the uptake flue leading to the room above, and the higher the flue the stronger will be the flow of air.

Air filters or washers are very desirable in buildings of this kind on account of the damage done to goods by soot and dust. The washer occupies less space than a dry filter, is more effective, and requires less attention. When cloth screens are used a large area is necessary and provision should be made for removing the accumulation of dirt from the filtering surface by means of an attachment to the vacuum cleaning system.

20. Exhaust Ventilation.—It is not usual to provide exhaust ventilation from the main floor, as the various stairways and elevator shafts serve to distribute the entering air throughout the building, from which it passes outboard by leakage.

In basements, which are comparatively low and without windows, provision should be made for removing a considerable proportion of the air supplied, by means of special fans.

A typical basement lay-out corresponding to Fig. 9 is shown in Fig. 10. In this building the fans for ventilating both the basement and first floor are located in the sub-basement. The basement is warmed by a single main heater at the base of the uptake flue, and the warm air delivered at a slight angle downward near the outer walls.

Exhaust ventilation is along the other side of the room through registers near the floor. Air for the first floor is tempered by a main heater near the fan and re-heated by stacks at the bases of the various uptake flues as shown. Rotation is secured by opening doors in the distributing ducts as indicated.

Toilets, lunch rooms, etc., require the same treatment as already described in connection with office buildings.

21. Volume of Air Required.—This may be based either upon the number of occupants or a certain number of changes per hour. In a large department store, erected a short time ago, the main floor was furnished with 75,000 cubic feet of air per minute, which was intended to supply a maximum of 2,500 people with 18 cubic feet each. This volume of air gave five changes per hour

for the first floor and served to supply the upper stories with a considerable amount of ventilation at such times as the main floor was only partially filled. The basement of the building, which was used principally for receiving and shipping purposes,

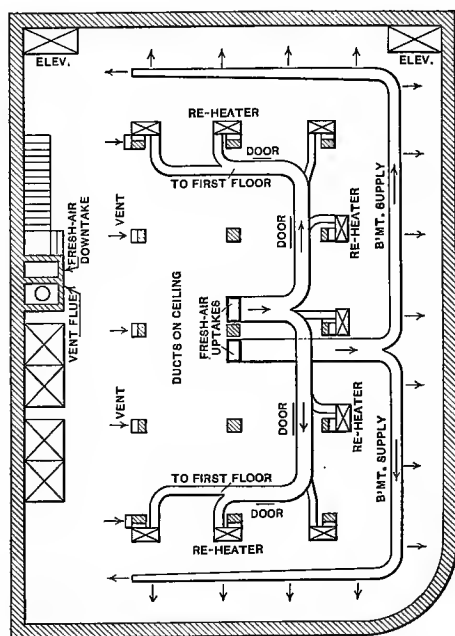


Fig. 10. Basement Ducts and Heaters.

was supplied with three changes per hour, which gave ample ventilation for the number of people regularly employed in this department.

22. *The Warming of Entrances.*—An important detail of store heating is the proper warming of the main vestibules. This should be done in such a manner as to reduce the entrance of cold air, when the doors are opened, to a minimum. The best results are brought about by introducing a comparatively large volume of warm air by means of a fan so as to maintain a considerable pressure between the two sets of doors. This has the effect of resisting a sudden inrush of cold air when the outer doors are opened and provides a supply of warm air which flows

into the store when the inner doors are used. The air is best introduced at each side of the vestibule near the floor, and the result is sometimes improved by adding one or more registers which discharge directly toward the outer doors. A typical arrangement of this kind is shown in Fig. 11. Waste heat is often utilized for this purpose by exhausting air from the basement at a comparatively high temperature, passing it through a shallow re-heater, and then discharging it into the vestibules as described above.

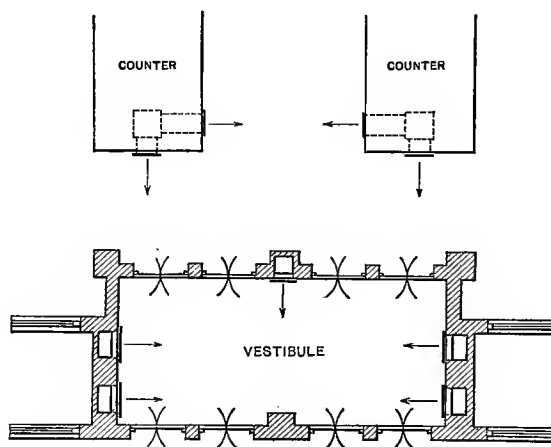


Fig. 11. Heating Arrangement for an Entrance Vestibule.

23. Arrangement of Direct Radiation.—This may be either in the form of cast-iron sectional radiators or wall coils as most convenient. Where pilasters project into the room between windows, and where the sills are low, good results may be obtained by the use of double return bend coils, as indicated in Fig. 12. When there is sufficient height beneath the windows, cast-iron radiators may be used if preferred, but if appearance is not important, wall coils are cheaper to install, and give a better distribution of heat. When the building is exposed to prevailing winds from certain directions, as the north and west, the cooling effect of inleakage may be offset by increasing the direct radiation on these sides of the building. The amount of increase is

usually taken as that required to warm a quantity of air equal to approximately one change per hour of the entire volume of the room. In determining the radiating surface, it is first computed in the usual manner for the entire floor, for the minimum outside temperature assumed, and evenly distributed on the outer walls. Correction is then made for leakage, as noted above, assuming the air to be raised from the minimum outside tempera-

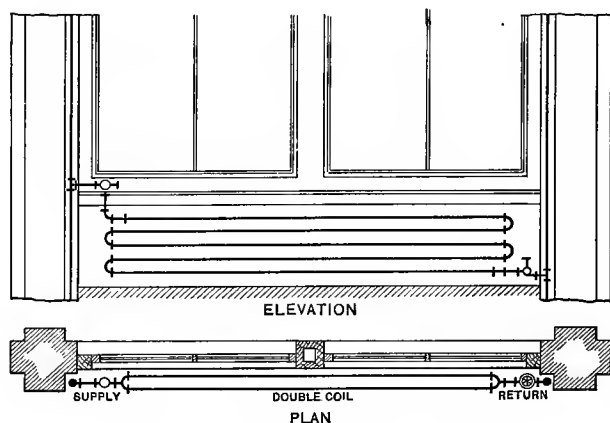


Fig. 12 Detail of Heating Coil beneath Window.

ture to 70 degrees. Another method is to compute the radiation for each side of the room separately and correct for the points of compass as described in the following chapter.

24. Systems of Heating.—The same systems of steam and water heating are adapted to this class of work as described for office buildings, and the same general arrangement of piping may be employed. When plumbing pipes are carried in the attic or roof space, the overhead system of steam or water distribution is to be preferred on account of the heat radiated from the mains and branches which prevents any danger of freezing in this location. Again, in buildings of this type, the basement or sub-basement ceiling is covered with pipes and ducts of various kinds and it simplifies matters to place the distributing system for the heating in the attic.

Municipal Buildings.

25. General Requirements.—This group includes such buildings as court houses, city halls, post-office buildings, public libraries, etc.

Buildings of this type are similar in their requirements to office buildings, with the addition of auditoriums, committee rooms, reading rooms, and others of a similar nature where a considerable number of people are gathered together for varying lengths of time. The smaller rooms are usually heated by direct radiation, the same as in office buildings, although, in this case, it is more common to furnish an independent air supply also. This is due to the limited number of stories, which makes it possible to reach the various rooms either with separate flues from the basement or from corridor airways as illustrated in Figs. 4 and 5.

26. Court Rooms and Auditoriums are best ventilated by means of a supply fan introducing the air through wall registers 7 or 8 feet from the floor.

Exhaust ventilation may be partly through ceiling vents and partly through grilles or registers near the floor. It is not usually necessary to provide exhaust fans under ordinary conditions, but if the available flue area is restricted, they should be furnished with sufficient capacity to remove at least 0.6 of the air supplied.

The location of both the supply and vent fans will depend somewhat upon the arrangement of the rooms. In general, it is best to place all ventilating equipment in the basement, thus making it more accessible, but when the room is at or near the top of the building, the fans may be located in the attic, or on the roof, in order to shorten the length of flues.

If rooms, ventilated by independent fans, were to be warmed only while occupied, it would be much simpler to do the entire work by means of the main heaters. As a matter of fact, it is desirable to maintain a comfortable temperature when ventilation is not required. Therefore, under these conditions, it is best to keep the two systems independent, heating by direct radiation and introducing the air at a temperature of 72 to 74 degrees, which is slightly above that of the rooms, in order to prevent down-drafts.

Sometimes it is possible to use supplementary stacks at the bases of the flues, which may be utilized as rotation heaters when the fan is not running.

This arrangement, however, is often difficult to carry out satisfactorily in practice owing to the building construction.

If there are high windows it is well to admit a portion of the air through narrow slots in the sills, placing re-heaters below, as shown in Fig. 13.

These stacks should be in addition to the direct radiation, as they have but little heating value, unless doors can be opened into the connecting duct to admit warm air from a basement room, when the fan is not running. Under this condition they may be counted the same as direct radiation in the room.

27. Committee Rooms and Offices.—These are best heated by direct radiation and may be ventilated by means of a special fan or connected with the main fan supplying the court room. While the duplication of equipment is to be avoided on the score

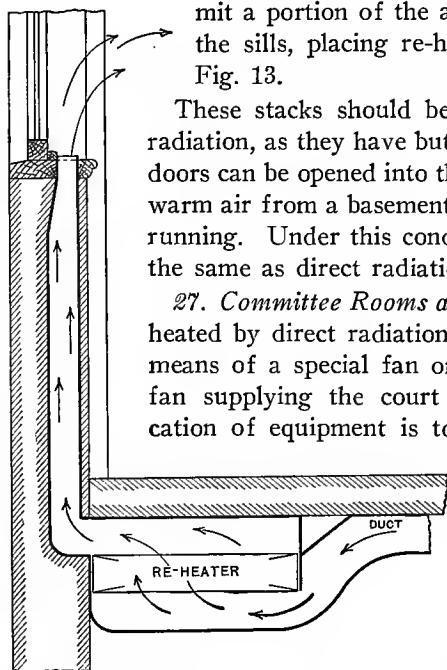


Fig. 13 Heater for Preventing Down-Drafts from Windows.

of economy, elaborate arrangements of switch dampers, and the changing of fan speeds and heating surfaces from time to time to meet varying requirements, complicates the operation of the plant and often results in impaired

efficiency. Hence, economy of first cost and convenience of operation should both be considered when deciding upon the most satisfactory arrangement.

Leakage may commonly be depended upon to remove the air from rooms of this kind, although in committee rooms, and others where sessions are likely to be held with closed doors and transoms, it is best to provide vent flues connecting with a shaft leading outboard. Exhaust fans are not usually required in cases

of this kind as the slight pressure within the room is sufficient to force the air out.

28. Post-Office Buildings.—The treatment here will depend somewhat upon the size of building. Ordinarily the smaller offices and corridors are heated by direct radiators, while the main work room is furnished with ventilation, either by means of a fan or indirect gravity radiation. When the latter arrangement is employed the stacks are usually made of sufficient size to heat the air supply to about 75 degrees in the coldest weather and direct radiation provided for independent heating. If preferred, ventilation and heating may be done by the same stacks, provided air

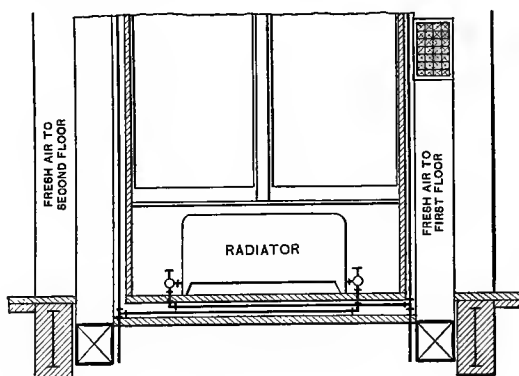


Fig. 14. Detail of Radiator Connections and Air Flues.

rotation can be arranged for when ventilation is not required. In all larger buildings, where the main work room is in use day and night, this provision is not necessary as ventilation will be required at all times.

29. Libraries.—Small buildings of this kind are usually heated by a combination of direct and indirect radiation, the latter furnishing ventilation for the more important rooms, such as reading and reference rooms, private offices, etc. In larger buildings a fan system should replace the indirect gravity stacks and the heating should be done independently. If the building contains an auditorium or lecture room it should be treated in a manner similar to that described for court rooms.

30. *Air Volume.*—This should be based upon the probable number of occupants, so far as possible, in the larger rooms, with four or five changes per hour in the smaller ones.

31. *Flue and Radiator Arrangements.*—A typical flue and radiator lay-out for this class of work is shown in Fig. 14. The radiator is placed beneath the window in a recess formed by pilasters at either side.

The supply flues connect with ducts at the basement ceiling and are carried up within the pilasters, together with the steam and return pipes, as illustrated in the cut. The run-outs to the radiators may be either in spaces beneath the tiling or above the floor, with provision for expansion as indicated.

Details already shown in connection with other buildings may also be adapted to this class of work.

Hotels.

32. *Rooms Specially Considered.*—The rooms requiring special treatment in the case of hotels and club houses are the dining and grill rooms, bar room, smoking room, billiard room, kitchen, toilets, and lavatories.

Buildings of this class are commonly heated by direct radiation, supplemented by re-heaters in connection with the ventilation of the more important rooms.

As this work does not differ essentially from that already described for other buildings under similar conditions, it need not be duplicated here.

33. *Main Lobby and Office.*—This, in buildings of small and medium size, is simply heated by direct radiation without making any attempt at artificial ventilation. In large hotels of the better class this room should be furnished with a generous supply of fresh air, and may, if desired, be warmed by increasing the size of main heater and combining the two processes in one. As buildings of this kind are in continuous use day and night the fan will always be running, hence, independent heaters are not required for this particular room, except under the special conditions mentioned later. If it is desired to reduce the amount of ventilation during the latter part of the night, the fan may be

slowed down so as to move just enough air to bring in the required amount of heat.

Exhaust ventilation is not commonly provided, the air being allowed to pass up the stairways and elevator shafts and find its way out by leakage from the upper floors. If, in any case, this room is rather low, it is well to provide a sufficient number of ceiling vents to carry off tobacco smoke, and even with high or vaulted ceilings, it may be advisable to assist leakage somewhat in order to prevent odors of any kind from passing to the rooms above.

34. Dining Room.—This requires positive ventilation, which is best secured by the use of fans. The heating may be done independently or it may be combined with the ventilating system as described for the main lobby. Sometimes the same fan is made to supply both rooms, being slowed down somewhat when the dining room is not in use. Each room, however, should have its own heater in this case, automatically controlled by a thermostat placed in the room. The air should first be drawn through a primary or tempering coil and its temperature raised to about 74 degrees and then blown through the supplementary heaters on its way to the rooms. When each room is provided with an independent ventilating unit, only one heater is required for each. This should be made up of several sections, part controlled by hand and part thermostatically. In general, the air is best introduced at an elevation, either in the window sills or through registers in the outer walls. When the system is used in summer, for cooling, separate inlets are sometimes provided near the floor which may be thrown into use by means of switch dampers. The reason for this is to avoid cool drafts from the falling air which are likely to occur when it is introduced from above. Exhaust ventilation should be partly at the floor and partly at the ceiling, to give a slight upward current in case there is smoking in the room. The air volume should be based upon the maximum seating capacity, allowing at least 40 cubic feet per minute per occupant. Grill rooms should be treated in a similar manner, either from the same or an independent system. Here there is likely to be more smoking and a much larger proportion of the exhaust should be through the ceiling. As the dining room is usually located directly above the kitchen and communicates with it through lifts and serving rooms, a

plenum condition should be maintained which will produce a continuous flow of air outward. This may be accomplished by exhausting artificially only about 60 per cent of the air supplied.

35. Bar and Smoking Rooms.—The air flow from these rooms should be strongly outward to prevent smoke and the odor of wines and liquors from passing into other parts of the building.

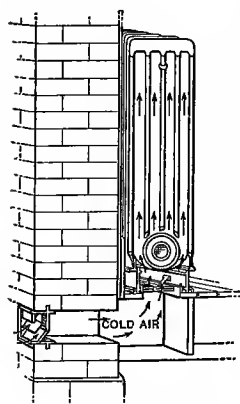


Fig. 15. Detail of Direct-Indirect Radiator.

A very satisfactory plan for rooms of this kind is to heat them by direct-indirect radiators, taking air from out of doors, and ventilating wholly by exhaust at or near the ceiling. By this arrangement there is no pressure within the room and the air movement is always toward the ceiling. Direct-indirect radiators, of the form shown in Fig. 15, are particularly adapted to these conditions because they not only furnish heat but also allow fresh air to be drawn in by suction to replace that exhausted at the ceiling. Rooms of this type should have at least eight complete changes of air per hour.

36. Billiard Rooms.—Rooms of this general character, which include lounging and reading rooms, may be heated in a similar manner to the main lobby. They should have from four to five changes of air per hour, and be arranged for exhaust ventilation with a part of the vent registers near the ceiling to remove the smoke. If isolated from other rooms it may be necessary to provide special fans for this section of the building, but it will usually be possible to combine these rooms with others, such as the main lobby or smoking room. When this is done it will be necessary to supply tempered air, and warm the rooms independently, either by direct or supplementary heaters.

37. Kitchen.—This should be provided with a strong exhaust ventilation in order to prevent any outward air movement to other parts of the building. The main part of the kitchen ventilation should be local rather than general. That is, it should be largely through hoods over the range, oven doors, vegetable steamers, soup-kettles, coffee urns, or any other piece of apparatus where smoke, steam or odors are likely to be generated. In addition to

this, there should be means for removing heated air from the upper part of the room when required, through openings controlled by dampers. The air supply may be partly from other rooms, such as serving room, servants' dining room, etc., and partly from out of doors through a special duct. Cool air is best admitted near the ceiling in front of the range through elbows which may be revolved to discharge in any direction desired. Ordinarily there will be enough heat at the upper part of the room to warm the entering air sufficiently to prevent uncomfort-

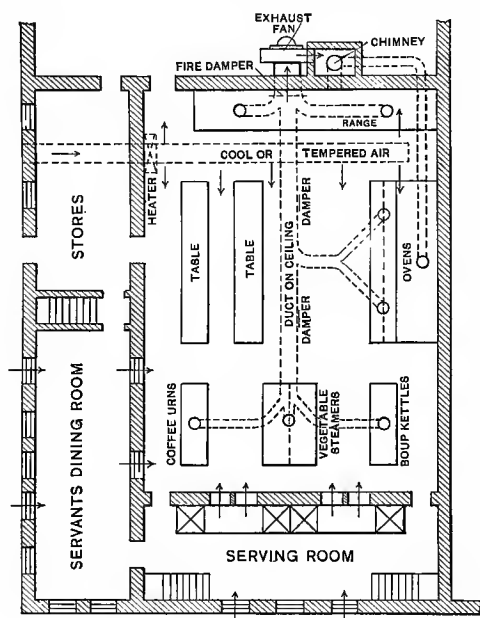


Fig. 16. Ventilation of a Large Kitchen.

able down-drafts, but in very cold weather, it is usually necessary to provide a heater for tempering it somewhat. This intake, as well as the ceiling vents, should be under control by means of adjustable dampers. In the largest hotels where it is not possible to secure the required amount of fresh air in this way, supply fans and tempering coils should be provided, taking care that the volume supplied does not exceed about 60 per cent of that ex-

hausted. A diagram illustrating the general scheme of ventilation for a kitchen is shown in Fig. 16.

Efficient hood ventilation depends upon the removal of air at a high velocity through a comparatively small opening rather than at a low velocity acting over a large area.

This condition may be secured in practice by constructing the hood as shown in Fig. 17, in which case the air is drawn partly through openings in the top of the hood and partly through a narrow slot, about an inch in width, extending around the entire perimeter.

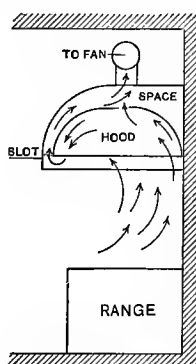


Fig. 17.

Fig. 17. Ventilating Hood for Range.

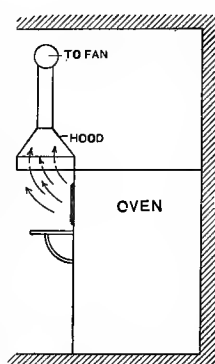


Fig. 18.

Fig. 18. Ventilating Hood for Oven.

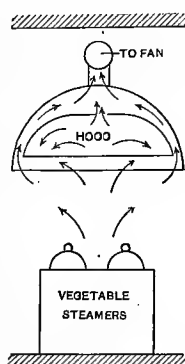


Fig. 19.

Fig. 19. Ventilating Hood for Vegetable Steamer.

Smoke and hot air rising from the range first strike the curved deflector, move forward as indicated by the arrows, and are caught by the strong suction at the edges of the openings and drawn into the vent flue. To secure the best results the velocity through these openings should not be less than 500 feet per minute.

Arrangements for the ventilation of ovens and vegetable steamers are shown in Figs. 18 and 19, respectively. In the case of an oven the greatest amount of heat is given off when the doors are opened, hence the hood should be placed directly above them to catch the outward rush of hot air before it can pass any distance into the room. In Fig. 19 it will be noticed that the slot

arrangement is made use of as previously described. Kitchens should be provided with fifteen to eighteen changes of air per hour if less than 14 feet in height.

Provision should first be made for securing the required velocity through the hoods, and ceiling vents furnished for the remainder of the work.

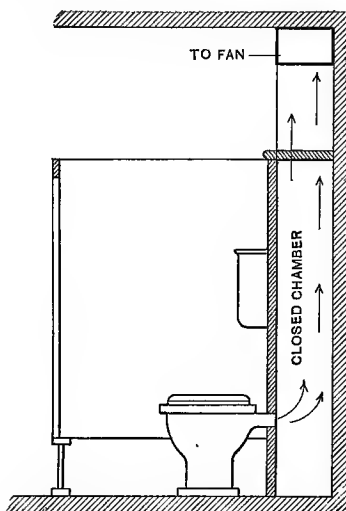


Fig. 20. Toilet Ventilation through Closed Chamber.

All ducts and flues beyond the range connections should be made fire-proof on account of the inflammable coating formed on the inside from the vaporized oils generated in cooking. When constructed of metal, No. 12 black iron should be used, insulated with block covering finished with cement. A fire damper, held open by a fusible link, should be provided to shut off the flue automatically in case of fire and thus check the draft. The same device should also be arranged to stop the fan motor at the same time.

38. Toilets and Lavatories.—Toilet rooms should be ventilated mainly by an exhaust through the fixtures. One of the most satisfactory arrangements is that shown in Fig. 20. Here a closed chamber of slate or marble is provided back of the seats and

urinals which also serves to conceal the various plumbing connections. Into this chamber are connected the local vents from the fixtures, which are usually from 3 to 4 inches in diameter.

A suction or exhaust is best produced by means of a fan as indicated in the cut. The velocity through local vents should not in general be over 200 or 250 feet per minute on account of drafts. After computing the air volume passing through the fixtures the remainder of the six or eight changes per hour may be provided for by general ventilation through adjustable registers. Room

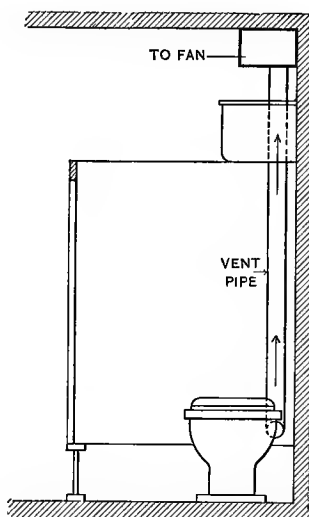


Fig. 21. Toilet Ventilation through Separate Pipes.

vents are best located near the floor and are often connected with the closed chamber back of the fixtures. When this is done, adjustments should be so made that they will not interfere with the assumed velocity through the local vents. A similar arrangement without the closed chamber is shown in Fig. 21. Lavatories are usually in the same room with the toilets and therefore do not require special treatment. When by themselves, general room ventilation is all that is required, placing the vents principally at the floor with a small proportion above the bowls to carry off any steam which may rise to the upper part of the room.

Hospitals and Institutions.

39. Systems of Ventilation Employed.—Until recently the most approved method of heating and ventilating hospitals has been by means of the closed system, so called, in which the windows are kept shut and all air introduced through indirect heaters either by means of a fan or by gravity. This system, when applied to city buildings, usually includes some form of air filter or washer, and special attention is also given to maintaining a uniform temperature day and night.

Within the past year or two investigations have been carried on which seem to indicate that while the above conditions are ideal, theoretically, better results are secured in practice by admitting a certain amount of fresh unheated air through the windows and by varying the temperature somewhat from time to time during the day.

Some well-known physicians even go so far as to recommend the abolishing of artificial ventilation, except for a certain amount of exhaust, and admitting the entire air supply through open windows.

While heating engineers are not ready, at present, to accept such radical changes, there seem to be good reasons for believing that a considerable amount of window ventilation may be made use of to advantage and that a certain temperature variation is desirable. Hospitals are particularly well adapted to window ventilation because the occupants of the wards, being in bed, are easily protected by extra blankets and by the use of screens, a condition which is not found in other buildings to any extent. Again, hospital patients occupying the same rooms continuously do not experience the changes in temperature which come to the average well person who is attending to his regular duties. Hence, special care should be taken to vary the temperature at different periods in order to secure this change artificially. If the wards are provided with automatic regulation the thermostats may be set to carry a somewhat lower temperature at night than during the day. Of course careful discrimination must be used in putting these methods into practice and the patient's condition should be considered in each individual case.

40. *Selecting a System.*—In view of the above, it would seem that in selecting a system of heating and ventilation for a hospital, provision should be made for both the “open” and “closed” methods, so they may be used alternately, or combined, as conditions seem to indicate. Such an arrangement is comparatively simple, the only requirement being to make the heating and ventilating systems independent, so that on mild days or at any time when it is desired to flush out the wards with fresh air, the fans or other source of air supply may be shut down, the windows opened, and the heat supply increased, if necessary, to maintain a comfortable temperature within the rooms.

41. *Types of Hospitals.*—Hospitals are of two general types, the large city institution consisting of buildings several stories in height, and those built on the cottage plan, usually found in the

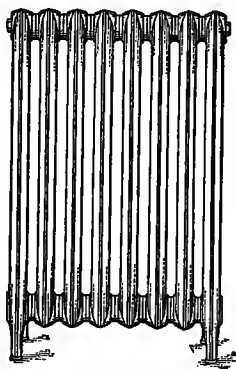


Fig. 22. Open Type of Direct Radiator for Hospitals.

suburbs and smaller cities where land is less expensive. These two classes of buildings require special treatment and will be taken up separately.

42. *City Institutions.*—The desired results are best obtained in this case by placing direct radiators in the rooms and supplying tempered air by means of a fan. Vent flues should be provided for use when operated as a closed system. The radiators should be of the open type, shown in Fig. 22 to facilitate cleaning. These are still better adapted to their work if the legs are omitted and

they are supported from the wall by brackets. Direct radiators used in this way should be of ample capacity and are best located beneath the windows in order that the air may be tempered somewhat as it enters. In the closed system, supplementary heaters are commonly used at the bases of the flues. These, however, have but little heating effect when the supply fan is shut down, thus making it necessary to rely upon direct surface when window ventilation is to be practised.

If there is ample flue space the exhaust ventilation may be under gravity circulation, assisted by an aspirating coil to prevent down-drafts when the supply fan is not running. Vent flues

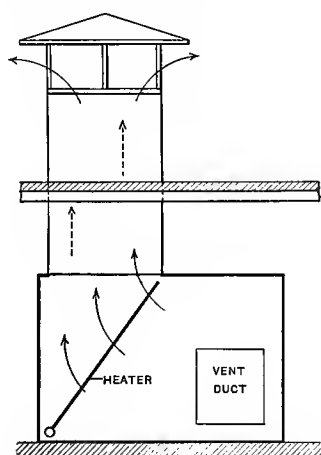


Fig. 23. Vent Chamber with Aspirating Heater.

from the various wards should be connected with one or more gathering chambers, each having a coil or heater placed in a manner similar to that shown in Fig. 23 and so arranged as not to reduce the area of the main shaft.

When the flues are necessarily of small size, or contain numerous offsets, it is best to employ a fan rather than an aspirating coil. Fans of the disk type require but little power when used in this way and are positive in their action under all ordinary conditions.

With the arrangement shown in Fig. 24 the fan may be shut down and the air by-passed through the damper in cold weather when there is a strong natural draft. All dampers of this kind, which require more or less frequent manipulation, should be operated pneumatically from a central switchboard.

The supply and vent flues in buildings of this type are usually carried up in the corridor or inside walls, although division walls between rooms may be utilized if necessary to obtain the required space. Both of these arrangements are illustrated in Fig. 25. A typical lay-out for a three-story building is shown in diagram in Fig. 26. The heating is done by direct radiation and is therefore independent of the ventilating system. Fresh air may be supplied

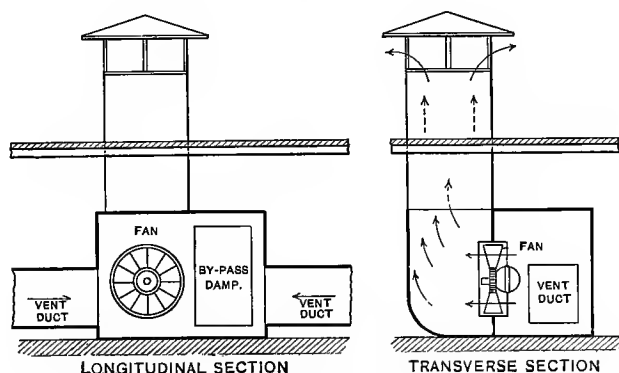


Fig. 24. Vent Chamber with Exhaust Fan.

either by a fan located in the basement or through open windows, as desired. Vent flues are carried from each room to a gathering chamber, from which the air is exhausted outboard by means of a fan provided with a by-pass as illustrated in Fig. 24.

43. *Cottage Hospitals.*—Institutions of this kind consist of a group of comparatively small buildings, usually one and two stories in height. Owing to their small size and the available flue space, it is not customary to employ fans in buildings of this type, except in connection with special rooms. There are two common methods of heating the wards, both of which are adapted to window ventilation when desired. In the first of these, direct radiators are used for warming, the same as in larger buildings, but

the tempered air is furnished by indirect gravity heaters instead of fans. When window ventilation is used the outside air supply to the stacks is shut off. In the second method the indirect stacks are made large enough for both heating and ventilation and the direct radiators are omitted. Provision should be made in this case for air rotation when the air supply to the heaters is shut off and window ventilation is in use. Discharge ventilation is practically the same in both cases, and through individual vent flues from the wards which are gathered into a central shaft in the attic. In one-story buildings the best results are obtained by placing loops of steam pipe in the flues leading from the smaller

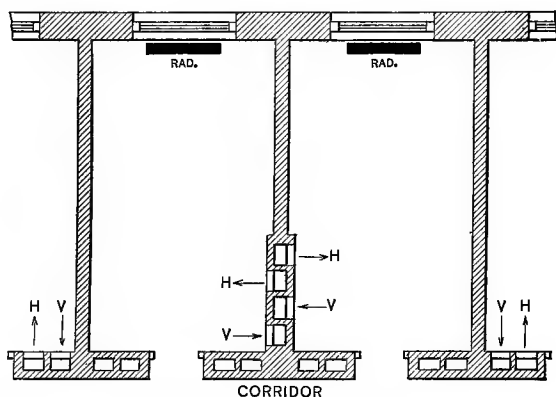


Fig. 25. Arrangement of Radiators and Flues for a Large Institution.

rooms, although a single coil in the main outboard shaft is sometimes used instead.

A typical section through the main ward of a one-story building, heated by direct radiation, is shown in Fig. 27. A very satisfactory arrangement for combined indirect heating and ventilating is illustrated in Fig. 28. In each case, the entire basement under the main ward is made use of for a cold-air chamber and the heater casings are constructed with open bottoms as indicated in the cuts. The warm air in Fig. 28 is admitted beneath the windows instead of at an elevation so as to temper the entering cold air when window ventilation is employed. Air rotation for warming at such times is secured by closing the cold-air inlets to

the basement and opening the rotation registers at the end of the ward which allow sufficient air to pass from the room into the basement air chamber to produce the necessary circulation through the heating stacks.

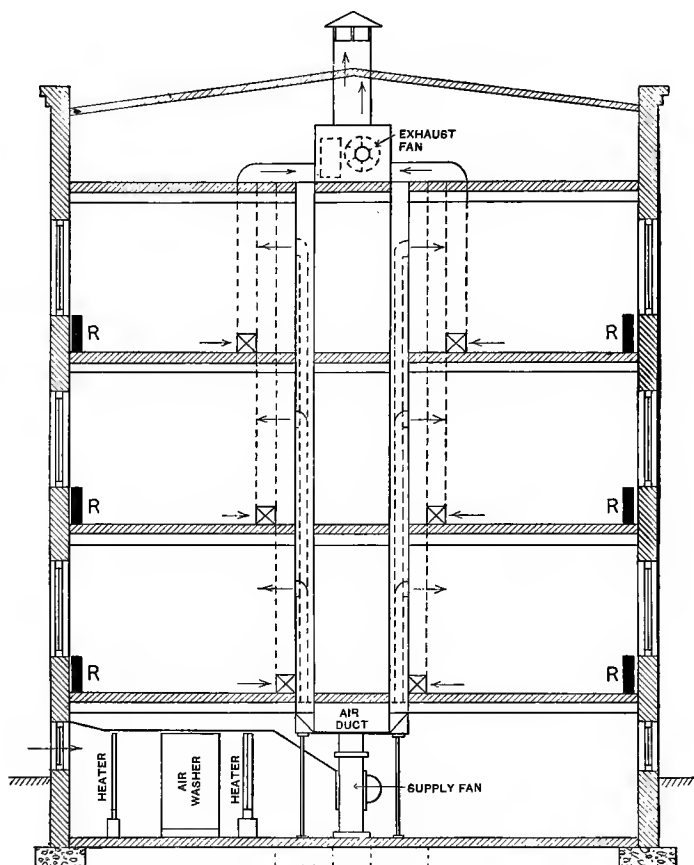


Fig. 26. Diagram of Heating and Ventilating System for a Large Hospital.

. In both of the methods described, the temperature of the entering air is regulated by mixing dampers at the stacks.

44. *Contagious Wards.*—These are always located in isolated buildings, but even under this condition care should be taken to cut off communication, as much as possible, between wards con-

taining different diseases. All supply ducts in a gravity system should be kept separate from each room back to the outer air, or at least to a trunk line well beyond the heater, in order to prevent dust being carried from one room to another by back-drafts through the ducts.

Vent flues should be run independently to the attic gathering chamber for the same reason. With a fan system there is not so much danger because the air currents are always in the same di-

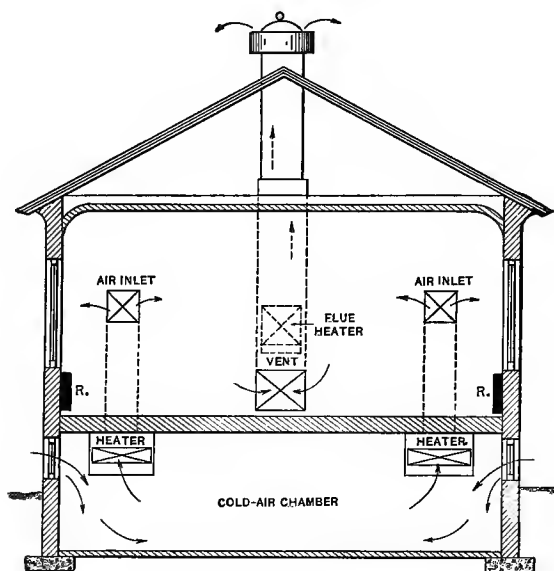


Fig. 27. Direct Heating and Ventilation for a Cottage Hospital.

rection when the fan is in use, although the same condition occurs in a measure when the equipment is shut down. This is guarded against in some cases by placing electrically controlled dampers in the ducts which close automatically when the fan stops.

45. *Operating and Etherizing Rooms.*—These rooms should be provided, in addition to the regular ventilating system, with special exhaust fans of small size under control of the physicians for clearing the air of ether fumes during or after an operation. Fans for this purpose should be of sufficient capacity to change

the air of the room in four or five minutes and should be arranged to draw from a point near the ceiling and discharge directly outboard.

Cool down-drafts from the large window above the operating table may be guarded against by placing two or more lines of steam pipe between the sashes. Operating rooms should be provided with sufficient heating capacity to raise the temperature to 80 degrees, if desired.

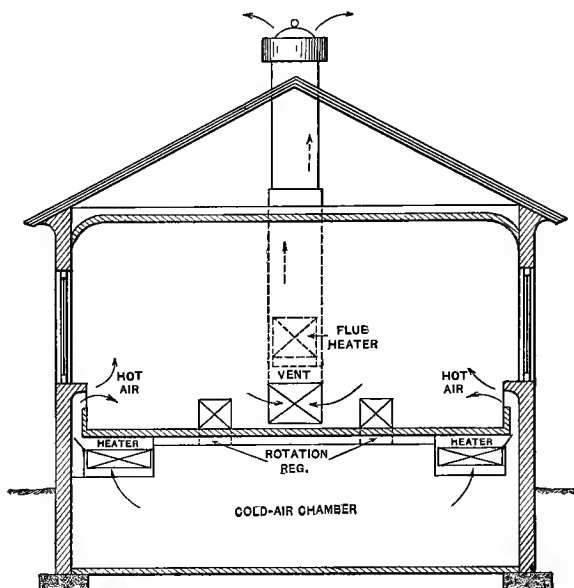


Fig. 28. Indirect Heating and Ventilation for a Cottage Hospital.

46. *Toilets and Service Rooms.*—Toilets, bath rooms, diet kitchens, serving rooms, etc., are best heated by direct radiation and provided with exhaust ventilation only. It is well in rooms of this kind to place the vent outlets at an elevation at least 7 feet above the floor in order to carry off any vapor which may rise from tubs, steam tables, or gas stoves.

47. *Corridors and Sun Rooms.*—These are commonly heated by direct surface in the form of circulation coils or radiators placed along the outer walls. Corridors require no special ventilation and open windows will usually suffice for the sun rooms.

48. *Laundry.*—This is an important department of a hospital or similar institution and should receive brief mention at this point. Unless a laundry is properly ventilated it becomes greatly over-heated, and as the air contains a high percentage of moisture the conditions are very enervating to those employed there. While open windows and roof ventilators may answer all purposes in warm weather, the free admission of cold air in the winter causes dangerous drafts and excessive condensation. One of the most effective methods of ventilation for rooms or buildings of this kind is shown in diagram in Fig. 29, and consists in the removal of warm moist air from the upper part of the room

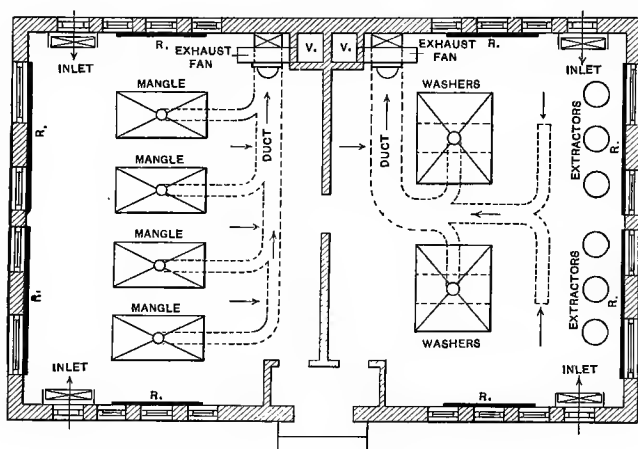


Fig. 29. System of Heating and Ventilation for a Laundry.

and the admission of fresh tempered air at one or more inlets to take its place. Heat and moisture should be exhausted, so far as possible, from the points where they are generated, and the best results are obtained by placing hoods over the washers, mangles, etc., and connecting them with an exhaust fan discharging outboard. A certain amount of general room ventilation should also be provided by means of ceiling ducts and registers as indicated. The special form of hood construction shown in Fig. 17 will not be required in this case as any steam which passes into the room will be carried off by the ceiling vents above

mentioned. The entering air is tempered by passing through shallow heaters consisting of cast-iron sections, of the pin or other similar type, which give ample space between them. These heaters should be made up of separately valved groups so that the temperature of the air may be easily regulated. Provision should also be provided for closing off the inlets by means of adjustable dampers or shutters. Direct radiators or coils are required beneath the windows to offset the effect of leakage and heat transmission and to warm the room when the ventilating equipment is not in use.

49. *Air Volume for Hospitals.*—The air supply for the wards should be based upon the number of occupants, allowing at least 80 cubic feet per minute each, in general and private wards, and 100 cubic feet in contagious wards. Toilets, bath rooms, diet kitchens, etc., should have from five to six changes per hour, and laundries from ten to twelve changes.

50. *Heating by Steam and Hot Water.*—Both steam, and hot water under forced circulation, are adapted to hospital work. Steam is the simpler for all forms of indirect heating, where temperature regulation is secured by mixing or by-pass dampers. It also has the advantage that danger of freezing in closed sections is avoided. Hot water on the other hand, is more easily regulated, unless automatic control is provided, and is therefore especially adapted to direct heating, both in large buildings and in groups of smaller ones.

Theaters.

51. *Arrangement of Ventilating System.*—The general scheme followed in the heating and ventilation of large modern theatres makes use of fans for both supply and exhaust, while the warming is done partly by indirect and partly by direct radiation. The main auditorium usually has but little outside exposure, so after it is once warmed, the animal heat from the audience is generally sufficient to maintain the desired temperature even in the coldest weather.

52. *Supply Ventilation.*—The most satisfactory method of supplying fresh air to rooms of this kind is to introduce the greater part at a low velocity through a large number of small openings

beneath the chairs. A common arrangement of air distribution is illustrated in Fig. 30 which represents a section through a theatre or opera house of large size.

Plenum spaces are provided under the floors of the main auditorium and balconies and connected with one or more supply fans, usually located beneath the entrance where the basement has its greatest height. The air is best delivered to these spaces through several ducts having flaring mouthpieces provided with vanes for

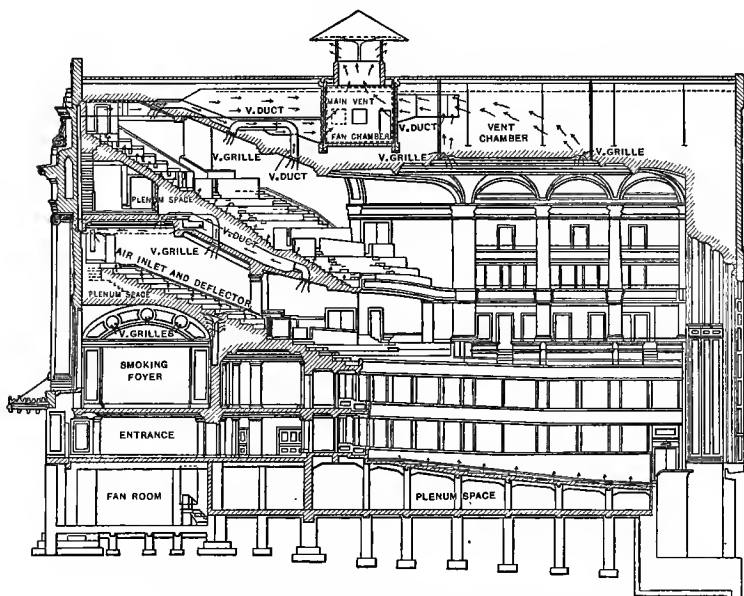


Fig. 30. Ventilating System for a Large Opera House.

diffusing the air and thus equalizing the pressure as much as possible.

The fans may usually be made to discharge directly into the lower space, while flues concealed in the side walls or central columns carry it to the balconies. The air is admitted through the floor in different ways, but usually by means of "mushroom" ventilators placed beneath the chairs, or in the spaces between them. The general form of this type of inlet is shown in Fig. 31, and is such as to deflect the entering air downward, thus

tending to diffuse it without sensible drafts. The same result is sometimes secured by means of perforated chair legs, of the form shown in Fig. 32. Private boxes should each be supplied with fresh air through special galvanized iron flues run in the walls or wherever the building construction permits. Flues of this kind should connect with special ducts leading back to the fan. If made to take their supply from the plenum spaces there will not be sufficient air pressure to produce the required velocity within them.

Foyers, waiting rooms, dressing rooms, offices, etc., are usually supplied by the main fan, although a separate one is sometimes used on account of complications in temperature regulation.

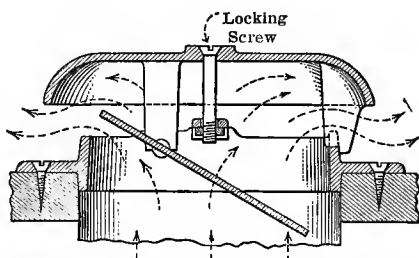


Fig. 31. Section through a "Mushroom" Ventilator.

53. *Exhaust Ventilation.*—The exhaust or discharge ventilation should be partly through the ceiling of the main auditorium and partly through the low ceilings above the balconies, as indicated in Fig. 30. The flow should be especially strong through the latter vents else the air will become pocketed in these low spaces and the balconies be overheated.

The ceiling vents are connected in the roof space with a centrally located chamber from which the air is discharged outboard by means of exhaust fans.

Smoking rooms and toilets, which require especially strong exhaust ventilation, are best provided with a separate fan and may be treated in the same general way as described in connection with hotels.

54. *Air Volume.*—The air volume for a theater is commonly based upon the number of occupants, allowing at least 25 cubic feet per minute each. This is somewhat less than is provided in schoolhouses and similar buildings, partly on account of the large cubic space per occupant, but principally because the air movement is continuously upward and the impure air passes directly from the breathing line to the ceiling vents.

Foyers, waiting rooms, dressing rooms, etc., should have from six to eight changes per hour, and smoking rooms and toilets, ten to twelve changes.

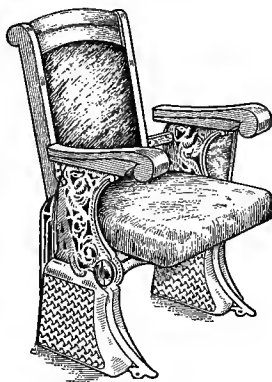


Fig. 32. Ventilating Chair
for Theaters.

55. *Heating.*—The auditorium is warmed principally by the general air supply, which is commonly brought in at a temperature of 65 to 75 degrees, but usually below 70 degrees when occupied.

Outside wall exposures are provided for by direct radiation, which may be in the form of screened or recessed radiators, or pipe coils, according to location. Foyers and waiting rooms are warmed both by supplementary heaters and direct radiators. When the latter are used in passageways they should be placed in recesses so as not to obstruct the floor space. Dressing rooms are best heated by direct radiation independently of the ventilation.

56. *Temperature Regulation.*—The temperature of theatres and auditoriums is best regulated by two thermostats, one placed in the room and set to maintain a uniform temperature of 68 or 70 degrees, and another in the duct or plenum space set to prevent the entering air from falling below 65 degrees, regardless of the room temperature.

Air cooler than this will prove uncomfortable when admitted near the feet and should be guarded against in this manner. The thermostats above mentioned are connected with pneumatic valves or mixing dampers and control the main heater at the fan. Other rooms are controlled independently by means of valves upon the supplementary or direct radiators. The air to these rooms may usually be taken from the main supply because its temperature rarely exceeds 70 degrees under ordinary conditions.

The main heater in all cases should be arranged for air rotation for heating the auditorium quickly without ventilation.

Technical and Trade Schools.

57. *Special Requirements.*—Next to hospitals in their requirements for efficient methods of heating and ventilation are the school buildings of different types. Technical and trade schools are included in the present list, because they often generate their own power, and therefore, present problems similar to those found in other large buildings combining power and heating in the same plant. So far as general classroom ventilation is concerned the methods employed in buildings of this kind do not differ from those in the ordinary city high school. In addition, however, technical schools contain rooms which require special treatment, such as machine and wood-working shops, laboratories, foundries, etc., thus making the problem somewhat more complicated in its details.

58. *Ventilation.*—What has already been said in connection with hospital ventilation applies in a certain way to schools. While drafts or air currents do not appear to be harmful, but rather the reverse when at the room temperature, it is not advisable to allow cool air to strike directly upon the occupants of a room unless they are especially protected against it by suitable

clothing. This may be carried out to a certain extent in hospitals, but except in mild weather, window ventilation in school buildings is usually confined to a thorough flushing out of the rooms during recess and the noon hour.

Here, as in most of the other buildings mentioned, it is necessary to keep the heating and ventilating systems separate in order to control the temperature in the different rooms. While supplementary heaters at the bases of the supply flues are still used to a considerable extent, direct radiation seems to be growing in favor for this class of work and has been used in many of the larger buildings recently erected. This makes it possible to introduce the fresh air at the normal temperature of the room, which seems to be a matter of considerable importance in maintaining its out-door quality.

59. Air Volume.—The air supply to buildings of this type should, in general, be based upon 50 cubic feet per minute per occupant for class rooms, recitation rooms, laboratories, etc. Corridors and recreation rooms should have at least four changes per hour. In the case of forge shops and foundries, which are often of considerable height, sufficient air should be supplied to make five or six changes per hour over a zone 12 feet in height.

Toilet rooms and lavatories may be treated the same as in hotels and office buildings.

60. Air Supply and Removal.—There are two general systems of air distribution employed in buildings of this kind, one of which is shown in diagram in Fig. 33. In this arrangement, which is especially adapted to buildings not over three stories in height, the fresh air distributing ducts are run either underground or at the basement ceiling and the vertical flues carried up in banks to the different floors as shown.

The main supply duct is commonly made to follow the line of the corridors with branches leading to the various uptakes.

An even distribution to the different rooms is secured by means of adjustable deflectors at the junctions of the branches with the main duct, supplemented by dampers in the vertical flues.

Discharge ventilation is due to the slight pressure within the rooms produced by the action of the fan and no exhausters or aspirating coils are usually required. The vent flues are car-

ried up separately to the attic, where they are joined into main shafts extending through the roof as indicated. The outboard vents should be provided with hoods, and dampers placed beneath them are necessary for shutting off the flow of air when school is not in session.

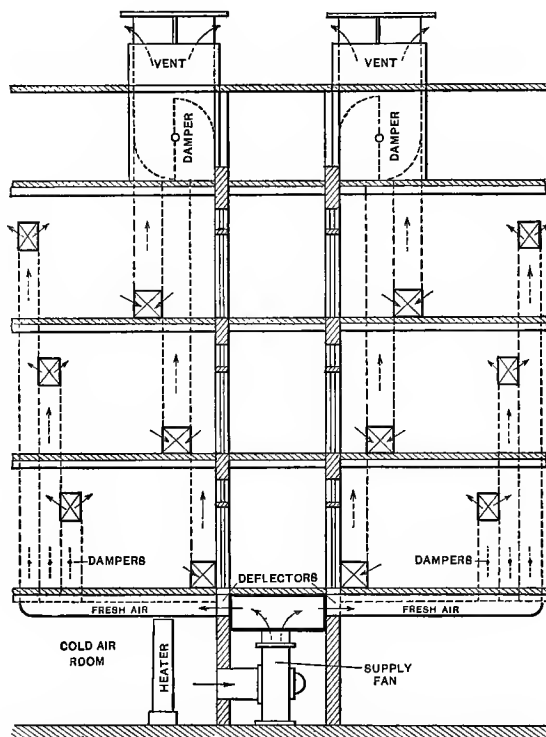


Fig. 33. Diagram of Ventilating System for a Large School Building.

The first floor plan of a small building employing this method of ventilation is shown in Fig. 34, and illustrates the general arrangement of the flues. Either direct radiation or supplementary heaters may be used with this system, as desired.

In the example shown, the class and recitation rooms employ the latter method of heating, with direct radiators at the en-

trances. Foot-warmers are placed in the main corridor near the centre as indicated.

The other general method of air distribution referred to is illustrated diagrammatically in Fig. 35. This arrangement is especially adapted to comparatively high buildings and where it is desired to eliminate the large distributing ducts from the basement. It is similar to the scheme already described in connection with office buildings and consists of a main distributing duct on the corridor ceiling of each story, supplied by an independent flue from the basement fan. Corresponding vent ducts are run parallel with the supplies and are connected with an exhaust fan usually located at the top of the building. An even air distribution is secured by means of regulating dampers which are permanently adjusted when the plant is installed. Connections between the corridor ducts and the rooms are illustrated

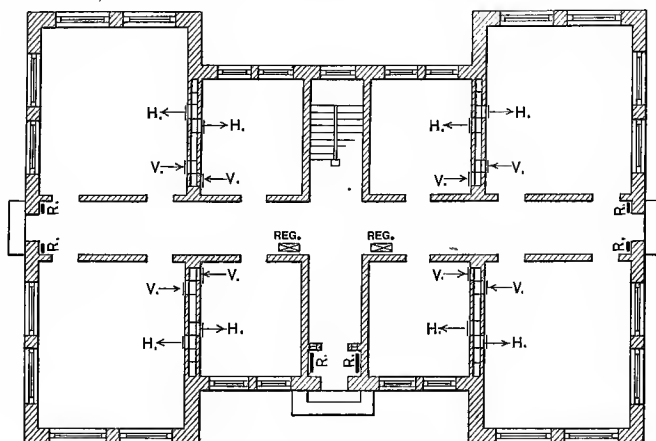


Fig. 34. Flue Arrangement for a School Building.

in Fig. 36. The first-floor plan of a building employing this arrangement is shown in Fig. 37.

This scheme of ventilation can only be used to advantage with a system of direct heating as it offers no practical opportunity for the installation of supplementary heaters. The diagrams shown in Figs. 34 and 37 are for illustrating principles rather than details of construction.

The same general arrangements may be enlarged to meet almost any condition and are often combined to advantage in buildings of large size. Although omitted from the diagrams, filters and air washes form an important detail in the equipment of large city schools.

61. *Heating.*—When supplementary heaters are used at the bases of the flues provision should be made for admitting inside

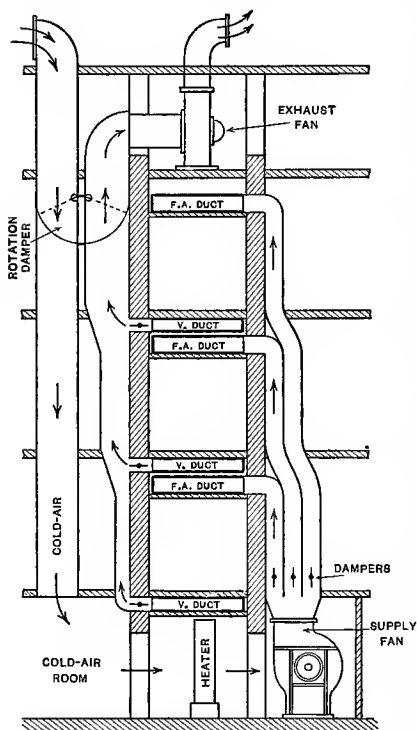


Fig. 35. Diagram of Ventilating System for High Buildings.

The advantages claimed for direct-indirect heating are the elimination of direct surface from the rooms, compactness of the piping system, and the avoidance of over-heating in the case of those sitting near the coil and radiators.

Direct heating, in buildings of this kind, has the advantage of distributing the heat where most needed, along the outer

air to the main duct for gravity warming when the fan is not running. Temperature regulation is secured either by attaching pneumatic valves to the supplementary heaters, actuated by thermostats in the rooms, or by the use of mixing dampers which by-pass a sufficient amount of tempered air around the heaters to give the desired temperature. With the first method the amount of heat stored in the stack is often sufficient to raise the temperature of the room somewhat after steam is shut off, and for this reason the best results are usually obtained by the use of mixing dampers operated by graduated thermostats which proportion the heat supply to the actual requirements at all times.

walls and beneath the windows, and requires no shifting of dampers or opening of doors for keeping the building warm when the fans are shut down.

Temperature regulation is best secured in this case by placing pneumatic valves upon the direct radiators and coils. In both systems the fresh air supply is maintained at a constant temperature by means of a duct thermostat attached to one or more sections of the main heater, or to a by-pass damper leading around it.

Air rotation, for quick warming, should always be provided when possible. With the arrangement shown in Fig. 33 a door opening into the cold-air room back of the heater, and connecting with the main corridors, will answer the purpose if the class room doors are left open to complete the circuit.

In Fig. 35 rotation is secured by means of a pair of rotation dampers connecting the cold-air downtake with the vent uptake at the third-floor level. When these are thrown, as indicated by the dotted lines, air passes from the rooms through the vent registers in the usual manner, and then, instead of passing to the exhaust fan, is returned to the cold-air chamber.

In general, class rooms, drawing rooms, laboratories, etc., should be maintained at a temperature of 68 to 70 degrees, while 62 to 64 degrees is sufficient for corridors, manual training rooms, shops, assembly halls and toilets.

62. Lecture Rooms.—These are usually provided with raised seats having a space beneath. In cases of this kind the best results are obtained by discharging the air supply into this space by means of a diffusing outlet and admitting it to the room through mushroom ventilators beneath the chairs as in theatres.

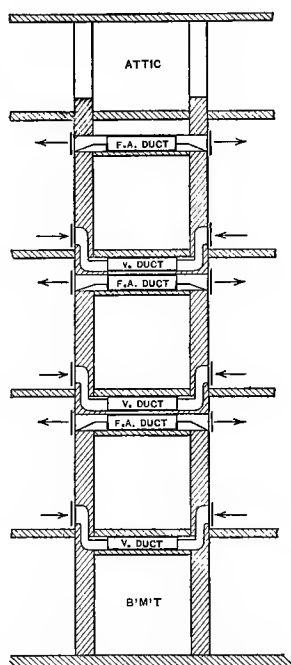


Fig. 36. Detail of Corridor Air Distribution.

Exhaust ventilation may be the same as for class rooms, that is, through wall registers near the floor at the sides of the speaker's platform. A good arrangement for heating a room of this kind is to place a double coil at the rear, beneath the raised floor, with a line of narrow register faces above.

In the case of interior rooms, lighted from above, sufficient heat may usually be furnished by the use of steam coils carried around the chamber between the upper and lower sashes. A typical lay-out for the heating and ventilation of a lecture room is shown in Fig. 38.

63. *Gymnasium*.—Many large city high schools contain a gymnasium which should receive careful attention when laying

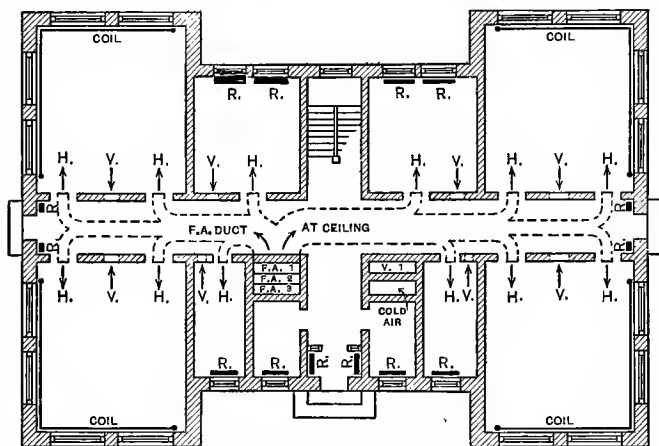


Fig. 37. Typical Floor Plan with Air Distribution at the Ceiling.

out the general system of heating and ventilation for the building.

Rooms of this kind should be furnished with an abundant air supply while in use. This is best admitted at a height of about 8 feet and exhausted from a point near the floor. When there is an elevated running track a portion of the air may be admitted 7 or 8 feet above it in order to distribute the supply as much as possible. Investigations show that a temperature of about 60 degrees, and a relative humidity of 45 to 50 per cent are the most satisfactory for rooms of this kind.

As a gymnasium is in use for only a portion of the time it should be arranged for independent operation by means of separate air ducts and lines of piping. Sometimes in large and important buildings it is provided with special fans and heaters.

64. *Laboratories and Kitchens.*—Chemical laboratories should be supplied with fresh air in the same manner as a standard class room, but a considerable portion of the discharge ventilation should be through the hoods or other exhaust openings under a strong suction.

Chemical hoods are most effectively ventilated when constructed on the general principle already shown in Fig. 17. A hood of this type, adapted to chemical work, is illustrated in Fig. 39. In certain processes a down-draft through the top of

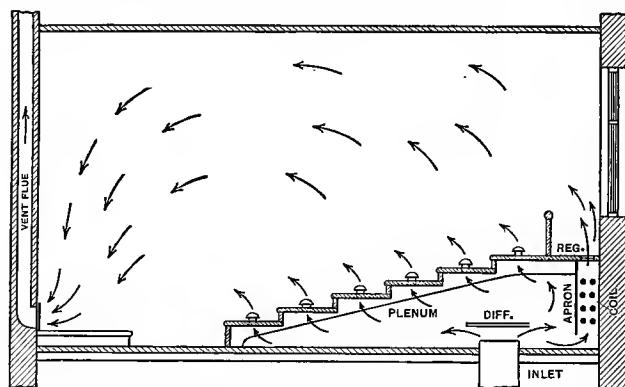


Fig. 38. Air Distribution and Heating Coil for a Lecture Room.

the desk or table is preferable to a hood, this being especially true in case of the instructor's table where a hood would obstruct the view from the seats. All hoods and draft outlets containing chemical fumes should be connected by means of tile pipe with a special exhauster so constructed as to resist corrosion. In laying out the vent ducts the amount of air necessary to produce the required velocity through the hoods should first be computed and the remainder of the discharge turned into the regular system, as general or room ventilation, through a register near the floor. In physical and electrical laboratories the heating pipes should be of brass to avoid the magnetic effect of iron.

Kitchen ventilation, in connection with lunch rooms and cooking departments, may be treated in a similar manner to that already described for hotels, although the requirements will be much lighter owing to the general character of the work performed.

65. Toilet and Locker Rooms.—The toilets are an important item in schoolhouse ventilation and should be treated in the general manner already described for other public buildings. In the case of trade schools, when the boilers are in constant use for power purposes even during the non-heating season, the toilets may sometimes be vented into a brick shaft surrounding the iron chimney-stack, thus doing away with special exhaust fans. If the stack is of brick or the power plant located in a separate building, this arrangement cannot be carried out.

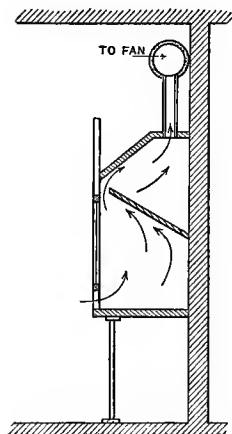


Fig. 39. Method of Ventilating a Chemical Hood.

A typical method of locker ventilation is shown in Fig. 40, in which case each individual closet is connected with a duct leading to an exhaust fan. Sometimes a line of steam-piping is run through the base of the lockers in order to facilitate the drying of wet clothing.

66. Shops and Foundries.—Woodworking and machine shops are usually treated in the same general manner as ordinary class rooms, except for local exhaust ventilation in connection with special machines, such as grinders, polishers, wood-planers, etc. In the case of forge shops and foundries the air is best admitted and discharged near the floor, a good arrangement being to place the supplies along the outer walls and the vents near the center of the room. Roof ventilation in foundries is not effective because the gases to be removed are heavier than air, and also on account of the amount of steam present which is rapidly condensed by the entrance of cold air through a monitor.

In forge shops a considerable proportion of the exhaust ventilation is through the forges, but general room ventilation is also advisable as noted above. Monitor openings should be pro-

vided for summer use and for such other times as it may be desired to ventilate by open windows.

67. *Steam and Water Heating.*—Both steam and hot water are suitable for warming buildings of this kind. Sometimes a combination of the two is employed, steam being used in the main heater at the fan, and hot water under forced circulation, in the direct radiation. Such an arrangement is especially adapted to buildings where automatic control is not installed.

Loft Buildings.

68. *Principal Uses.*—These are included in a class of buildings used principally for mercantile and light manufacturing purposes.

The varied character of the work and the frequent changing

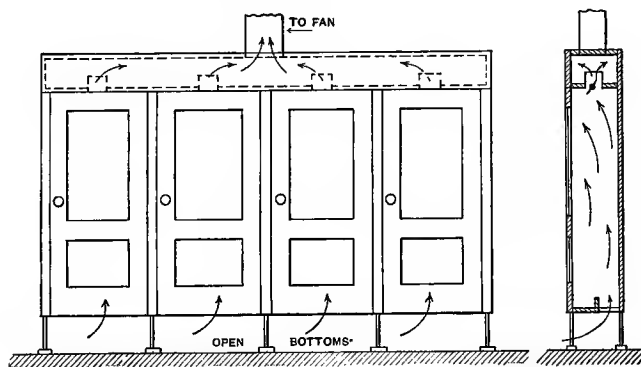


Fig. 40. Ventilation of Lockers.

of tenants usually makes it impossible to install a permanent system of ventilation as may be done in case of a store or factory building. Under ordinary conditions the owner provides heat, elevator service, and sprinkler equipment, together with the necessary power for their operation.

Electricity is commonly generated upon the premises and sold to the tenants for power and lighting, and in connection with this, both live and exhaust steam are supplied on each floor for manufacturing or other purposes.

69. *Heating.*—Buildings of this type are commonly heated by direct steam, the coils or radiators being placed along the outer walls beneath the windows. Automatic temperature regulation is not usually provided, partly on account of the expense and partly because the rooms, being large, usually contain several radiators thus giving a certain amount of hand control. In some cases, when the floors are not divided by partitions, unit hot-blast equipments are provided of sufficient capacity to do a portion of the heating, thus reducing the amount of direct radiating surface required.

This, however, can only be done to advantage when the future use of a given floor is to be permanent, or at least, is to remain the same for several years. As buildings of this kind are not generally heated during the night it is necessary to provide means for quick warming in the morning, which makes it advisable to employ some form of vacuum system for producing a rapid circulation through the radiators.

When the coils and radiators are of large size the best results are obtained by connecting a vacuum pump with the main return. Otherwise the single-pipe system with a vacuum air line will usually prove satisfactory. In laying out the piping special supply and return risers should be carried to the top of the building, with outlets on each floor, to provide steam for ventilation, should this be furnished later.

70. *Ventilation.*—The ventilating equipment in loft buildings is usually installed by the tenants in order that it may be adapted to their special needs. There are two common methods employed, one being to force in tempered air, allowing it to find its way out by leakage or through open windows, while the other makes use of an exhaust fan which produces a slight vacuum within the room, thus causing fresh air to enter through induction heaters placed in front of windows in different parts of the room.

The first method is usually favored by engineers, as being more easily controlled and giving a better air distribution when properly installed. In order to obtain the best results, however, provision must be made for the outward flow of air either through elevator shafts and stairways or by opening the windows slightly upon the leeward side of the building.

A typical lay-out for the heating and ventilation of a single floor in a loft building is shown in Fig. 41. The room is warmed by wall radiators placed beneath the windows and the temperature controlled by means of hand valves.

The ventilating equipment consists of a steel plate blower

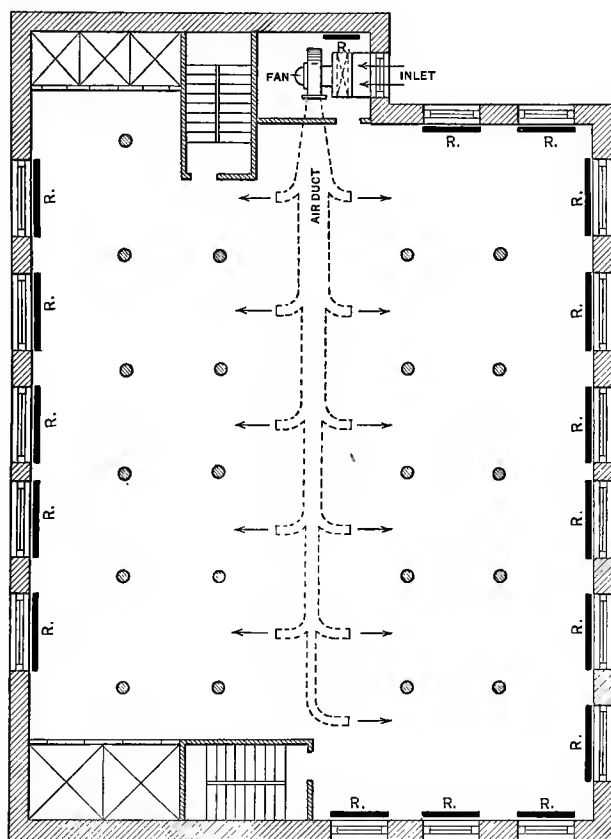


Fig. 41. System of Heating and Ventilation for a Loft Building.

driven by an electric motor and connected with an encased heater of sufficient capacity to warm the entering air to a maximum of 74 degrees in the coldest weather. The fan and heater are mounted upon a platform suspended from the ceiling of the toilet

room, and therefore do not require space which may be utilized for other purposes. The air distribution is through a galvanized iron duct at the ceiling with outlets discharging toward the outer walls.

When the space is divided by partitions the ducts must be laid out in such a manner as to give each room its proportion of fresh air. Rooms devoted to manufacturing processes requiring special conditions as to temperature and humidity must be heated independently.

When the exhaust method of ventilation is made use of, instead of the plenum, it is evident that provision must be made for supplying an amount of tempered air equal to that discharged by the fan.

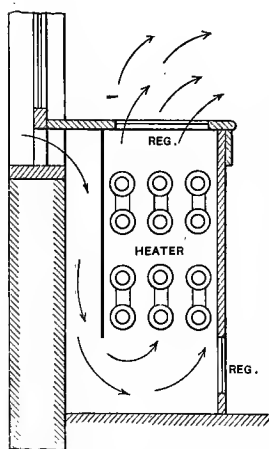


Fig. 42. Section through an Induction Heater.

An arrangement for this purpose is shown in section in Fig. 42, and consists of a double bank of indirect radiator-sections, placed before a window, and so encased that the entering air must pass through them before reaching the room. The volume of air admitted is governed by the height to which the sash is raised or a special adjustable damper may be placed in the inlet opening if preferred.

Temperature regulation may be secured by valving the heater in sections or by the use of a mixing damper similar to that shown in Fig. 7. During such times as ventilation is not required the window may be closed and the register in

front of the casing, near the floor, opened to produce a rotation of air through the heater. A sufficient number of these induction stacks should be provided to supply the required amount of tempered air, and they should be so placed as to diffuse it as much as possible.

Buildings of this type are commonly furnished with 25 to 30 cubic feet of air per occupant per minute.

Railroad Stations.

71. *General Requirements.*—The large railroad station or terminal, presents a rather complex problem, as it takes on the requirements of a number of buildings combined in one. The power plant must include equipment for the generation of steam, electricity, gas, compressed air, and a refrigerating outfit for the manufacture of ice, while the building itself contains offices, waiting rooms, restaurants, kitchens, barber shops, smoking rooms, toilets and lavatories.

Much of this work has already been covered in connection with office buildings and hotels and attention will only be given here to rooms not included in the above.

72. *Waiting Room.*—A detail of especial importance in equipping a building of this kind is the heating and ventilating of the main waiting room. It is best, in rooms of this kind, to eliminate all direct radiation on the score of cleanliness and combine the equipment for both heating and ventilation in a single system.

When a special fan is provided for the waiting room the entire heating may be done by a single main stack or coil, divided into separately valved sections and provided with a mixing or by-pass damper for the final temperature regulation. Rooms of this character are in constant use day and night so that the fan must be run continuously, therefore, gravity rotation for night warming is not necessary.

Under these conditions the single unit method above described is the simplest in application, especially when automatic temperature regulation is employed for controlling the by-pass damper at the fan. When the heating and ventilation of the public waiting room is combined with that of others, such as private waiting rooms, restaurants, barber shops, etc., the air should only be raised to about 65 degrees at the main coil, further heat being supplied by supplementary stacks placed in the individual flues. This arrangement is necessary in order to secure independent temperature control for the different rooms.

A typical lay-out for a public waiting room is shown in Fig. 43, and illustrates an effective method of introducing the fresh air and removing the foul air from rooms of this kind. The outer wall beneath the windows is thickened sufficiently to conceal

the flues, and the fresh air is brought in through long and rather narrow grilles at an elevation of about 8 feet above the floor. Exhaust ventilation is through similar grilles placed in the base-board between the inlets, as indicated.

The main supply and vent ducts connecting with the fans are carried directly beneath the flues in the basement.

The relative locations of inlet and outlet grilles are practically the same whether the warming is done by a single main heater or by supplementary stacks in the flues.

A section through one of the stacks in the latter arrangement is shown at the left in Fig. 44, which indicates the positions of the fresh air and vent ducts and also the connections with the heater.

A similar section through one of the vent outlets is shown at the right in the same cut.

Either steam, or hot water under forced circulation, may be

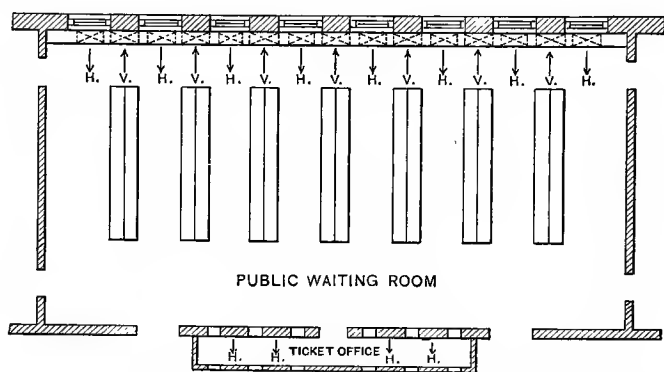


Fig. 43. System of Heating and Ventilation for a Public Waiting Room.

used in the direct and supplementary heaters, as seems best under existing conditions.

A vacuum system attached to the main return is generally employed when steam is used, in which case, a fair degree of temperature control may be secured by means of a throttle or graduated valve placed in the supply line leading to the supplementary heaters, and operated by hand. In the case of a hot-water system regulation may be obtained by varying the temperature of the water circulated through the heaters.

For more accurate control in either case a pneumatic system should be employed.

The air supply in rooms of this kind is commonly based upon 30 to 40 cubic feet per occupant per minute.

73. *Other Rooms.*—Baggage and express rooms should be supplied with direct radiators or coils of sufficient capacity to maintain a temperature of 50 degrees in the coldest weather.

These are best located near outside doors in order to temper the entering air to some extent. Small offices in connection with baggage rooms should be warmed to 68 or 70 degrees.

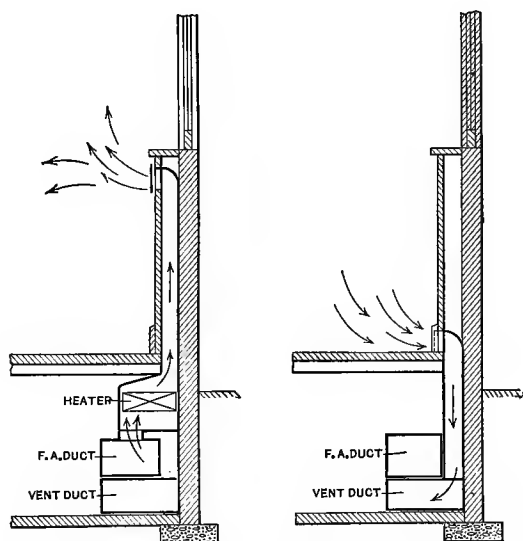


Fig. 44. Details of Supply and Vent Ducts for Waiting Room.

The toilets may be treated as already described for hotels, the only difference being the larger capacity of the equipment.

Machine Shops.

74. *Systems of Heating and Ventilation Employed.*—Both direct radiation and fan systems are adapted to the warming of machine shops under proper conditions. Buildings of this

class are practically free from steam and disagreeable or harmful gases except in special departments, and the cubic space is usually large compared with the number of occupants. Under these conditions leakage may be depended upon largely for supplying the fresh air for ventilation.

In small buildings, and large ones, when they are divided into stories from 10 to 14 feet in height, very satisfactory results may be obtained by the use of direct steam coils carried along the outer walls beneath the windows, especially if enclosed in such a manner as to ensure a proper circulation of air over them.

This method of heating is also adapted to cases where the plant is made up of a group of small or medium size buildings instead of one or two large ones, as the duplication of fan and heater outfits would prove expensive. Again, direct radiation is convenient for heating special rooms or departments requiring different temperatures from the main shop, such as offices, drafting rooms, shipping rooms, paint shops, etc.

Generally, in large plants a combination of the two systems is required to give the best results.

The disadvantages of direct radiation include the lack of ventilation, except by leakage, and also, when used in rooms having a large glass exposure, such an amount of surface is often required as to make it uncomfortably warm for those working near the coils. This is especially true in buildings of modern construction, having considerable height, with galleries and monitor roof. In this case the warm air rises rapidly to the upper part of the building, greatly over-heating it and increasing the rate of transmission through the roof. At the same time the zone near the floor occupied by the workmen remains too cold for comfort, except in the immediate vicinity of the radiation.

Sometimes coils are suspended from the roof trusses for heating the upper part of the building, so that only a nominal amount of surface is required at the floor. This corrects, to a certain extent, the fault of local over-heating, but is wasteful in the use of steam owing to the high temperature of the roof space as already noted. Direct radiation also produces but a small amount of circulation within the room, so that fresh air which finds its way in by leakage is poorly distributed.

In large buildings of modern construction, the best results are obtained by the use of a fan system when the apparatus is properly designed and installed. With this arrangement, the warm air may be so distributed as to heat the space evenly, and at the same time set up a circulation which will carry the in-leaking fresh air to all parts of the building.

75. *Heating by Direct Radiation.*—A typical arrangement of the heating coils in a one-story building with monitor roof is shown in Fig. 45. In this case part of the surface is placed at an elevation and part along the outer walls beneath the windows.

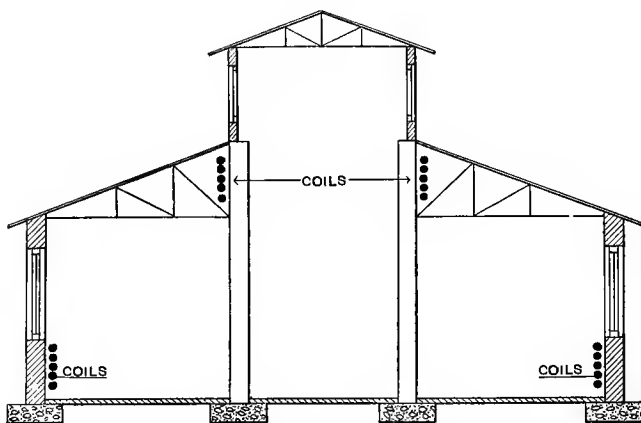


Fig. 45. System of Direct Heating for a Machine Shop.

An excellent plan for a wall coil is shown in Fig. 46, in which case the bench is set out about 4 inches and an apron of galvanized iron carried down on the rear supports to a point slightly below the lower pipe. With this arrangement, the warm air is not thrown out from beneath the bench into the workman's face, as with the usual method, but passes upward in front of the window where heat is most needed. By properly proportioning the height of bench and window, and raising the sash slightly, a good supply of fresh air may be obtained without unpleasant drafts.

Both steam and hot water under forced circulation, are adapted to this class of work; although when the latter is used the

pumps must be kept moving slowly on very cold nights in order to prevent freezing in the more exposed pipes.

As the heating mains in plants of this kind are of considerable length, and the radiating surface more or less extended, a vacuum system attached to the return is usually employed in the case of low-pressure steam to accelerate the circulation when first warming up in the morning and to avoid the use of large pipes and sharp grades in connection with the return system.

This also makes it possible to employ graduated supply valves upon the coils and radiators, so that the temperature may be controlled by hand, with fairly good results. This may be simplified by connecting up the radiation in sections in such a manner

that one valve will control the entire surface in a single room, or upon one side, in those of large size.

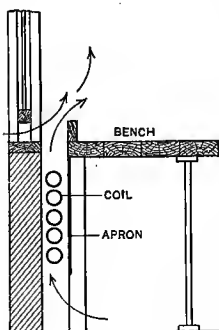


Fig. 46. Detail of Wall Coil.

76. Hot-Blast Heating.—The advantages of this system have already been described to some extent. The heater may be supplied with steam at any pressure desired, but is usually made to utilize the exhaust the same as in direct heating. Temperature regulation may be secured in different ways, the most common being to shut off a part of the sections or by-pass a portion of the air around the heater. If there is an ample supply of exhaust steam, the best method is to admit a

larger proportion of fresh air as the outside temperature rises.

When the system is arranged for the re-circulation of air, return ducts should be provided with inlets from different parts of the building. Otherwise the rotation of air will be localized in the vicinity of the fan and the circulation in other parts will be either weak or entirely lacking.

The best location for the apparatus will depend upon the general construction of the building and other local considerations.

A plan commonly adopted in shops of brick and steel construction is shown in diagram in Fig. 47. In this arrangement the

apparatus is placed overhead and the main distributing ducts carried through the roof trusses as indicated.

From here branches are brought down on posts and the air discharged directly downward, or at a slight angle, the greater portion being thrown toward the outer walls.

The openings should be from 5 to 6 feet from the floor, and the horizontal distance from the outer walls should not be more than 20 feet to get the best results. By this means it is possible

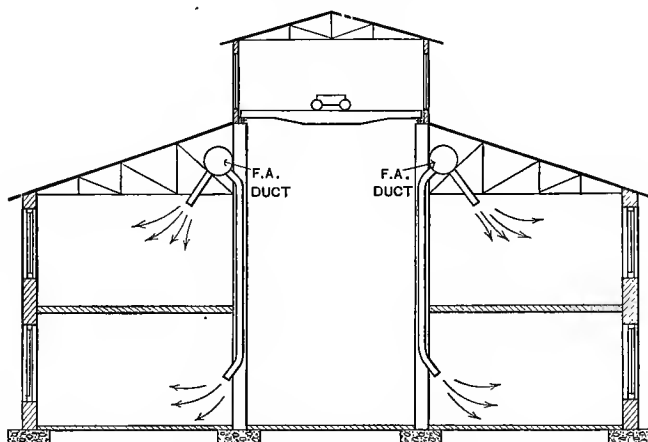


Fig. 47. Hot-Blast Heating System for a Machine Shop.

to maintain a stratum of warm air near the floor, where it is most needed.

In buildings having several stories, ranging from 10 to 14 feet or more in height, the air is often blown downward at an angle of 30 or 40 degrees from the horizontal, as in Fig. 48.

With smooth ceilings, even with a low air velocity, the heat will be evenly distributed for a considerable distance away, and with a return or vent duct at the center, good results are obtained for distances up to 100 feet or more. For buildings not over 60 feet in width, a good arrangement is to carry up a central flue or stand-pipe every 80 or 100 feet and discharge the air to each story through four outlets as indicated in Fig. 49.

Return ducts for bringing the air back to the fan for re-circulation, are commonly carried along the outer walls, and may be

either underground as shown at the right in Fig. 50, or overhead as shown at the left. In either case the inlets leading to them should be near the floor.

77. Temperature Required.—In machine shops, where most of the occupants are actively engaged, a normal temperature of 60 to 65 degrees is considered ample, while the offices and drafting rooms should be maintained at 68 to 70 degrees in the coldest weather.

It often happens that while the lowest continued temperature

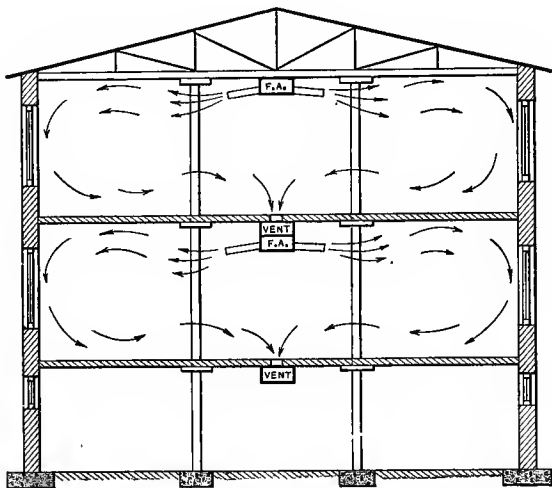


Fig. 48. Hot-Blast System for a Shop of Several Stories.

does not go below +10 degrees, or possibly zero, there may be infrequent periods when it drops to -10 or -20 degrees for a day or so. The question, therefore, arises, should the apparatus be designed for this lower temperature or only for normal conditions.

Experience seems to show that systems proportioned for zero temperature may be made to answer fairly well for short periods of considerably colder weather, and at the same time prove less wasteful under the conditions of average weather throughout the heating season. For example, a shop designed for a temperature of 60 degrees in zero weather will drop to 52

degrees when it is -10 degrees outside, which will not prove uncomfortable in the majority of cases.

Offices and drafting rooms should never be allowed to fall below 66 to 68 degrees and the radiation should be proportioned accordingly.

78. *Air Volume*.—The air volume to be handled by the fan is usually based upon one of two assumptions. In the first of these, which is the more common in the type of building shown in Figs. 47 and 50, the plant is designed for heating entirely with re-circulated air when the outside temperature drops to zero, and the proportion of outside air gradually increased as the tem-

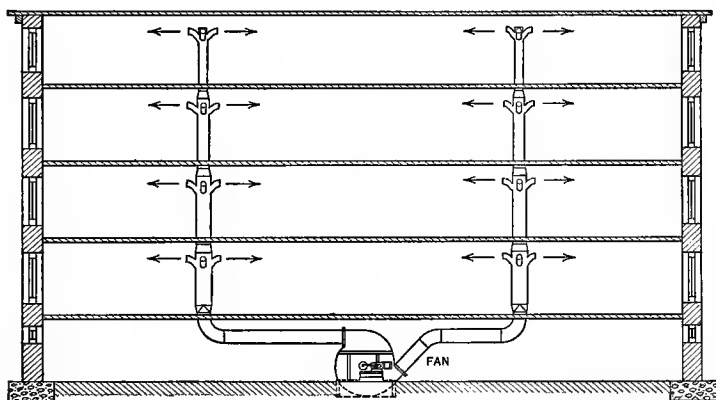


Fig. 49. Stand-pipe System of Hot-blast Heating.

perature rises. In this case the volume of air circulated must be sufficient to transmit the necessary heat for warming the building with a temperature range of 80 to 90 degrees as described in Paragraph 137.

When there is an abundance of exhaust steam, or the number of workmen large compared with the cubic contents of the building, a volume of outside air equal to one to three complete changes per hour should be supplied in the coldest weather, increasing this to 30 or 40 cubic feet per occupant per minute as the outside temperature rises.

Foundries.

79. *Fan Arrangement.*—In the case of foundries a fan system is always preferable, as here the air is filled with steam and heavy gases. Ventilation by means of skylights or monitor windows, except in warm weather, only serves to make conditions worse, as the entering cold air causes the steam to condense on the walls and overhead construction and drip to the floor. By means of a pair of fans, one for supply and one for exhaust, conditions may be greatly improved and the air of the building kept in good condition. In general, the air should be admitted and discharged near the floor; first, because warmth and fresh air are needed in the zone occupied by the workmen; and, second, be-

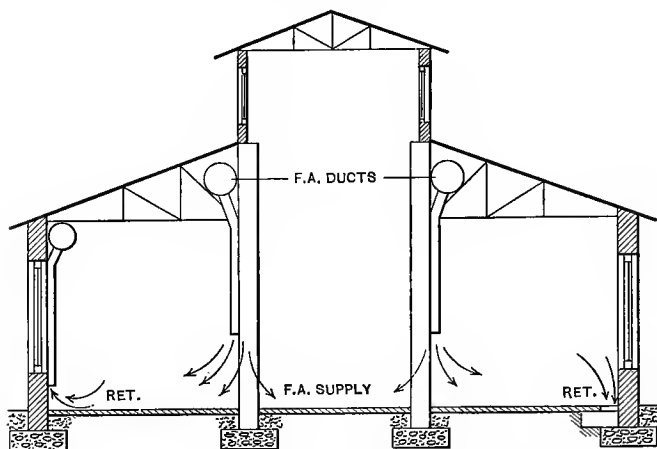


Fig. 50. System of Hot-blast Heating for Re-Circulation.

cause the gases present are heavier than air and must be drawn off through openings near the floor level.

When the work is of such a nature that steam and gases are present only in small amounts, the exhaust fan may be shut down and only the supply fan used.

A typical arrangement for the ducts and outlets in a foundry or forge shop is shown in Fig. 51. With this method of distribution the smoke is kept from collecting under the eaves, and in summer time, when the exhaust fan is not running, the admis-

sion of fresh air at the outer walls tends to drive the smoke into the monitor, from which it may be discharged through the windows.

Under ordinary working conditions, the air, after admission gradually rises through the building, and as it cools, falls through the central portion and is drawn off at the exhaust outlets, thus carrying with it the heavier gases which are given off in the processes of foundry work.

A method of connecting with an underground vent duct through vertical registers, is shown in Fig. 52.

80. *Temperature and Air Volume.*—The heating equipment of forge shops and foundries is usually designed to maintain a temperature of 50 to 55 degrees in zero weather, the higher figure allowing for a drop to -10 degrees outside, without pro-

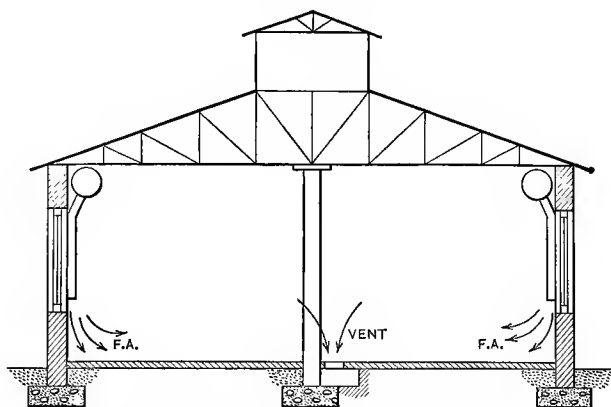


Fig. 51. System of Heating and Ventilation for a Foundry.

ducing an uncomfortably low temperature in the building for the character of the work carried on.

In the case of a foundry it is usually necessary to furnish a certain amount of fresh air at all times to clear the building of smoke and gases. The volume will depend somewhat upon the height, the required number of changes per hour being less, as the cubic contents increases, for a given floor area. An air supply equivalent to four or five changes per hour for a stratum 10 or 12 feet high over the entire floor should prove sufficient for ordinary conditions in iron foundries.

Brass foundries are more difficult to ventilate and should be provided with two or three times as much air as stated above. In designing the ventilating system for a building of this kind it is customary to make the supply and exhaust fans of the same size, regardless of the fact that a certain amount of air passes out by leakage.

Paper Mills.

81. *Ventilation of Machine Room.*—The machine room of a paper mill requires special methods of ventilation owing to the large amount of moisture given off by the drying paper, which amounts to approximately 1.5 tons per ton of finished product. If this moisture is not quickly removed, it produces a heavy fog, which rapidly condenses upon contact with the cool side walls

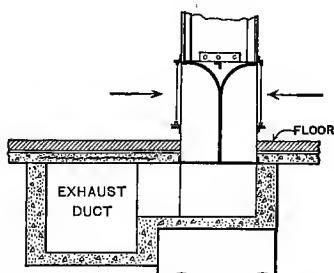


Fig. 52. Detail of Vent Outlet.

and roof, thus causing continuous dripping upon the operatives and upon the paper in process of manufacture. Furthermore, the high temperature and the extreme humidity are very enervating to those employed in rooms of this kind.

82. *Methods of Ventilation.*—While the various methods employed in the ventilation of paper machine rooms may vary in detail, the general principle is the same in each.

This consists in supplying a sufficient volume of warm air to absorb the moisture rising from the dryers, by evaporation, and to exhaust the air from the room before it becomes saturated.

Knowing the capacity of the machines, in tons per hour, the weight of water to be evaporated is easily determined, and also the volume of air necessary to carry it away under given con-

ditions. In practice the temperature of the entering air should not be less than 65 to 70 degrees, the actual point being governed by the heat required to maintain a temperature between 70 and 80 degrees in the room. The heat from the rolls is ordinarily sufficient to raise the temperature of the air rising from the dryers to 100 or 110 degrees, with the room temperatures above noted.

Computations for determining the required air volume under different conditions are given in Paragraph 162.

One method of ventilation is shown in diagram in Fig. 53, and makes use of a hood, having a slot from 2 to 3 inches in width around the perimeter, and extending well beyond the limits of the machine. This may be constructed of galvanized

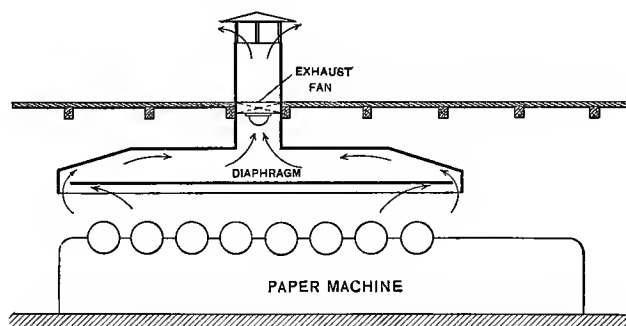


Fig. 53. Hood Ventilation for a Paper Machine.

iron with a deflector of stout cloth attached to removable wooden frames, or it may be made entirely of metal.

An exhaust shaft, located at the center, is provided with a fan of sufficient capacity for removing the required volume of air, as computed. Warm air should be supplied to the room in equal amount by a centrifugal fan, thus maintaining a balanced pressure which prevents the inleakage of cold air in any appreciable quantity. This may be admitted partly near the floor and partly in horizontal jets near the ceiling, the latter preventing to a large extent the gathering of moisture upon the underside of the roof.

The objection to a hood is that it shuts off the light somewhat and may also be in the way of the operatives if hung sufficiently low to secure the most effective results.

Some engineers prefer to omit the hood over the entire machine and substitute a number of fan-shaped exhaust outlets, directly above and surrounding the dryers, connected with a centrifugal fan. In addition to these, vent shafts are provided for general room ventilation, usually in connection with exhaust fans. In order to protect the roof construction, these shafts are often brought down some distance below the ceiling, so as to draw off the moist air before it comes in contact with the woodwork above. This action is still further assisted by the use of heated air jets blown across the ceiling at a high velocity. The balance of the air supply may be admitted near the floor in a manner similar to that already described for shops and foundries, care being taken to distribute it evenly throughout the room. In computing the heating surface for warming the air, the dryer rolls should be taken into account, allowing from one-fifth to one-

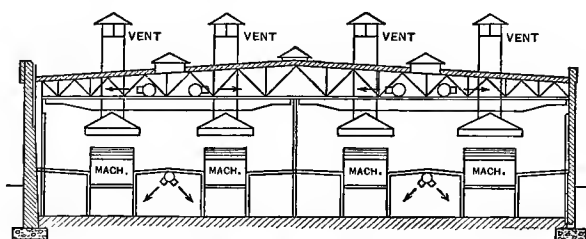


Fig. 54. Ventilation of a Paper Machine Room.

fourth of their surface for this purpose. Sometimes the buildings are so arranged that warm air may be drawn from other rooms, thus serving to ventilate them, and also utilizing a certain amount of heat which would otherwise be wasted.

A combination of the methods above described is illustrated in Fig. 54, which shows a cross-section through a double machine room. In this case, hoods are provided, but at a sufficient height to avoid any interference with the operation of the machines. Air to the main room is supplied horizontally near the ceiling in a sufficient quantity to give six or seven changes per hour. This is supplemented by two changes per hour delivered to the basement between the machines.

83. *Ventilation of Grinder Rooms.*—In pulp mills special ventilation is required in the grinding rooms, owing to the heavy

vapor which is given off from the logs as they are fed into the grinders. The best results are secured in rooms of this kind by arrangements similar to those already described for machine rooms, although hoods are not usually employed in this case. A typical system for the ventilation of a grinder room is shown in Fig. 55, in which sufficient warm air is supplied to produce from six to eight changes per hour. The equipment consists of a regular hot-blast outfit (not shown in the cut), having a heater of sufficient capacity to warm the air to a temperature of 125 to 130 degrees in zero weather. The distributing ducts are located

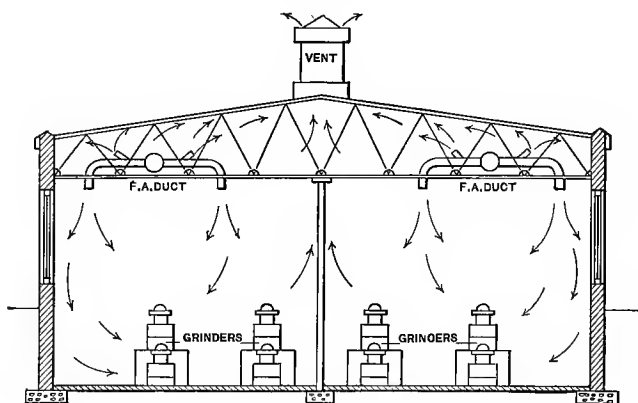


Fig. 55. Ventilation of a Grinder Room.

in the upper part of the room, and about three-fourths of the air supply discharged directly downward as indicated, the remainder being blown in thin jets along the ceiling in order to eliminate condensation as much as possible. Exhaust ventilation is through roof vents, and, in this case, is produced by the pressure within the room, without the use of fans.

Textile Mills.

84. General Requirements.—The general requirements for the satisfactory heating and ventilation of a textile mill consist of an even distribution of heat; an ample supply of fresh outside air, or purified air, if re-circulation is practiced; means for regulating the humidity of the various rooms, and an arrangement of the

equipment which shall neither obstruct the light nor the passages used by the operatives.

85. Systems of Heating.—Both direct radiation and hot-blast systems are used for work of this kind, either separately or in combination.

The former is adapted more especially to one-story buildings, having roofs of “saw-tooth” construction, where overhead airways tend to cut off the light and underground distribution is attended with excessive loss of heat, unless expensive insulation is provided.

While a direct system is less expensive to install and operate, from a purely heating standpoint, it does not furnish ventilation or provide means for regulating the humidity.

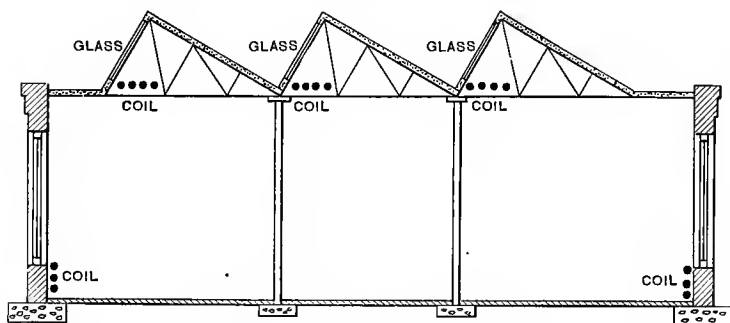


Fig. 56. Direct Heating for a Textile Mill.

Hot-blast heating, on the other hand, overcomes these objections but at an increased expense. By re-circulating the air in the colder weather the cost of operation may be reduced, but this makes it necessary to install washers and return ducts, thus increasing the original outlay.

When deciding upon a heating system for a textile mill it is necessary to consider the matter of humidity control at the same time, as the latter has an important bearing upon both the economy of installation and operation. For example, it may be found in a given case that a direct system will heat a building satisfactorily at a lower cost than a fan system, but that a separate equipment will be required to maintain the desired humidity.

On the other hand, a hot-blast system, taking its full supply of

air from out of doors, may be considerably more expensive than simple direct heating, but by combining it with an air washer, re-circulation may be practiced and the humidity controlled without additional equipment.

86. *Heating by Direct Radiation.*—Both hot-water, under forced circulation, and vacuum steam systems are used with good success in mill work. In the case of saw-tooth construction a large proportion of the heat transmission is through the roof, and it is customary to locate the greater part of the radiating surface directly beneath the glass, as shown in Fig. 56. This not only places the coils out of the way of machines and operatives, but tends to prevent excessive condensation upon the cold glass surface due to the high percentage of humidity maintained in the room. Sometimes all of the coils are suspended, but it

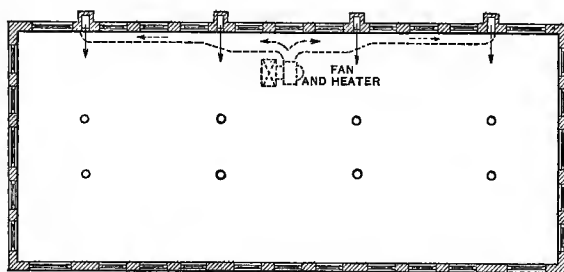


Fig. 57. System of Hot-blast Heating for a Textile Mill.

is better, if possible, to also carry a few lines of pipe along the side walls beneath the windows, as shown in the cut. Condensation is often guarded against still further by using double sash in the windows, with a line of pipe between them.

87. *Hot-Blast Heating.*—A typical lay-out for hot-blast heating is shown in Fig. 57. In this case the apparatus is located in the basement and a main supply duct, usually of masonry, carried along the outer wall, as indicated.

The uptake flues are run in pilasters, which project outward, in order not to encroach upon the passageway between the machines and wall. The warm air is all delivered at one side of the room and usually toward the colder or more exposed wall.

The smooth ceilings offer no special obstruction to the flow

of air and the moving belts and pulleys tend to break up the currents and assist in a more thorough diffusion.

A section through one of the uptake flues is shown in Fig. 58. These are spaced from 40 to 70 feet apart and deliver air to each story through openings placed about 8 feet from the floor. The flue area is reduced at each story and the volume of air admitted through the openings is regulated by means of an adjustable deflecting damper of special construction.

The main distributing duct is commonly made in the form of a quadrant and reduces gradually toward its extremities.

Heat loss should be guarded against, as much as possible, by air spaces or other means of insulation.

When the fan is driven from the line shafting a special motor or engine should be provided for use when the main engine is not running.

Another flue arrangement is shown in section and elevation in Fig. 59. In this case a separate flue is carried from the main air duct to each floor through piers between the windows. This has the advantage of avoiding outside pilasters, but increases the number of flues somewhat. With the piers 12 feet apart, it will be 36 feet between inlets in three-story buildings and 48 feet in those four stories in height.

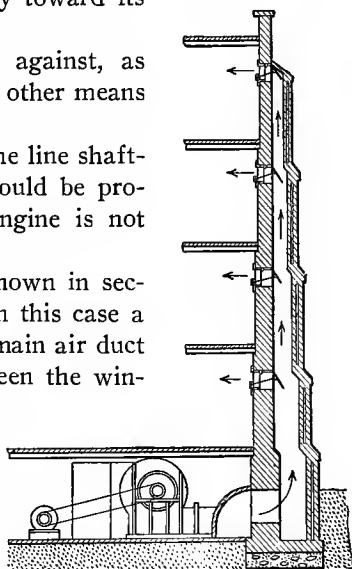


Fig. 58. Section through a Typical Uptake Flue.

88. *Humidity Control.*—Mention has already been made of the necessity for maintaining a certain degree of humidity in textile mills. This is due to the presence of frictional electricity caused by the motion of belts and machinery when the air becomes too dry. The effect of this in spinning rooms is to produce unevenness in the yarn, while it often results in a variation in the width and quality of woven goods. This condition may be practically overcome by maintaining a relative humidity of 60 to 70 per cent at the normal temperatures carried in buildings of this kind.

The air is most easily moistened when the hot-blast system is employed, and may be done by means of steam jets, evaporating pans or by the use of a spray washer or filter. The latter device is especially efficient in this case, because it also removes the dust and lint from the air and makes it fit for re-circulation in cold weather, which is a matter of considerable importance when the supply of exhaust steam is limited.

In some cases, however, it is possible to equalize conditions in different departments by drawing the air supply for a room, in

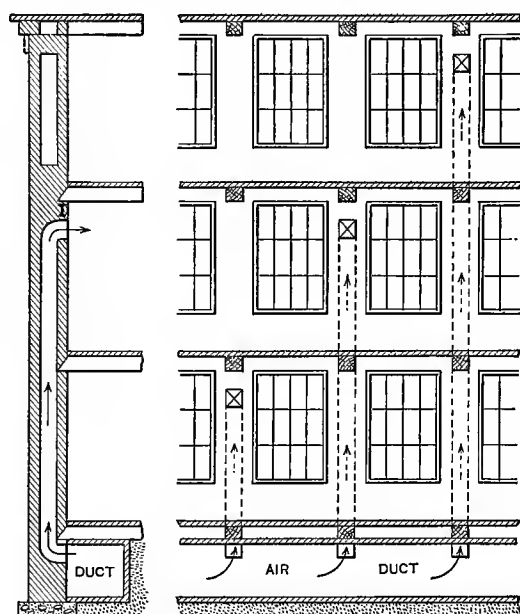


Fig. 59. Arrangement with Independent Uptake Flues.

which the humidity is too low, from one having too much moisture, thus gaining the desired result without extra expense.

89. *Ventilation*.—When there is an abundance of exhaust steam, the air supply should be taken from outside, unless special means are provided for removing the dust as noted above. In cases of this kind, at least four changes of air per hour should be provided, for the best results. When it is purified the amount

of outside air may be reduced, thus accomplishing a considerable saving in fuel.

90. *Ventilation of Dye Houses.*—Here the conditions are similar to those found in the machine room of a paper mill, it being necessary to remove large amounts of vapor without allowing it to condense upon walls and ceiling. The volume of air required will depend upon the amount of vapor given off, which is a maximum where the dye is kept at the boiling point and the cloth passed through it from reels located above the vats. It is common practice to provide for a complete change of air every two to ten

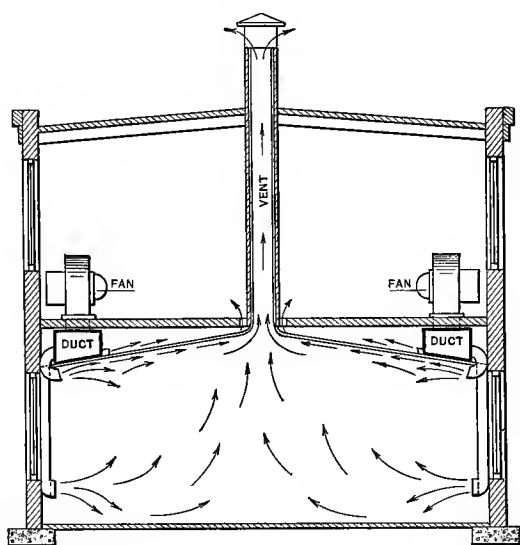


Fig. 60. System of Ventilation for a Dye House.

minutes according to the processes carried on in the room. A typical ventilating system for a dye house is shown in diagram in Fig. 60. This building is two stories in height, with the room to be ventilated on the lower floor. A false ceiling is provided, sloping upward from the outer walls to the vent flue at the center. Fresh warm air is furnished by two hot-blast outfits connecting with main distributing ducts carried in the furred space above the false ceiling. There are three sets of supply inlets as follows: First, those delivering air near the floor along the outer walls;

second, those blowing thin jets along the under side of the false ceiling; and third, those admitting a small amount into the furred space to prevent the gathering of moisture and the consequent rotting of the building construction. The general scheme of air movement is indicated by the arrows; that delivered near the floor supplying the operatives and carrying the vapor upward toward the vent stack, while the upper inlets furnish a thin sheet of warm air along the ceiling, thus preventing condensation at this point. The main vent shafts should not be more than 20 or 25 feet apart, and should be properly insulated and drained to prevent any dripping upon the fabrics in process of manufacture, or upon the employees,

CHAPTER II.

WORKING DATA FOR STEAM AND HOT-WATER HEATING.

The present chapter gives in condensed form working data for the design of steam and hot-water systems employing direct and indirect radiation with gravity air flow. This data is based upon constants and formulae widely used in practical work and has been arranged with special reference to the requirements of the buildings under consideration.

Heat Loss.

91. Classification of Buildings for Heat Loss.—The first step in the design of any system of warming, whatever its type, is to determine the heat loss from each room in B. T. U. per hour. This depends upon the character and extent of the exposed surface, the difference between the inside and outside temperatures, the points of compass, and exposure to winds.

The buildings described in the preceding chapter may be included in four general classes, according to their construction and heat requirements. Class I. includes office and municipal buildings, hotels, hospitals, theatres, schools, loft buildings, etc., where the walls are constructed of brick or stone with an inside furring of lath and plaster, and in which a normal temperature of 70 degrees is desired. Class II. includes shops and factories of brick or concrete, without furring, usually several stories in height, and also requiring a temperature of 70 degrees. Class III. includes shops of the general type shown in Fig. 47, having walls of brick or concrete and thin roofs of different construction. Buildings of this type are usually heated to 60 degrees. Class IV. includes foundries, boiler and forge shops, etc., covered entirely, or in part, with corrugated iron, and heated to 50 degrees.

These different classes of buildings will now be taken up in order, with tables and data for computing the heat loss from each within the usual range of conditions.

92. *Class I.*—For buildings in this class the following data may be used.

TABLE I.

HEAT LOSS THROUGH WALLS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 70 degrees.

Solid wall with lath and plaster furring and air space.	Heat loss in B. T. U.
8-inch brick	22.5
12 " "	17.5
16 " "	14.5
20 " "	12.0
24 " "	11
28 " "	10

For brick walls with sandstone face, 4 to 8 inches in thickness, plastered inside, but without furring or air space, multiply the factors in Table I. by 1.3. For stone wall instead of brick, multiply the factors in the table by 1.4

TABLE II.

HEAT LOSS THROUGH WINDOWS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 70 degrees.

Character of window.	Heat loss in B. T. U.
Single sash	84
Double sash	42
Single skylight	91
Double skylight	46

For other temperature differences than 70 degrees, multiply the factors in Tables I. and II. by the following:

TABLE III.

FACTORS FOR OTHER TEMPERATURE DIFFERENCES.

50 degrees, multiply by	0.71
60 " " " "	0.86
70 " " " "	1.00
80 " " " "	1.14
90 " " " "	1.29

The use of these factors will be found necessary in making computations in various parts of the country, as the outside temperature commonly runs from 10 to 30 degrees below zero in the

West and Northwest, while in the South it may pass through all points up to 60 or 70 degrees above, according to location.

In addition to the heat loss by transmission, there is also a certain amount due to leakage, depending upon the tightness of the building construction, especially around doors and windows, and upon the strength of prevailing winds. This may be provided for in the average structure of this class by multiplying the transmission loss by 1.2. In case of public rooms, where there is a good deal of passing in and out, it will be necessary to use a larger factor, depending upon local conditions. In main lobbies and similar rooms a factor of 1.5 to 2.0 is often required. Corrections for different points of compass should be made as follows:

TABLE IV.
FACTORS FOR EXPOSURE.

North exposure, multiply by	1.3
East " " "	1.1
South " " "	1.0
West " " "	1.2

In rooms of considerable height the heat loss is increased on account of the higher mean temperature and the more rapid circulation of the air over the cool surfaces. This amounts to about 2 per cent for each additional foot above 12 feet in height. Although in office buildings the walls are reduced in thickness on the upper floors, it is not usually necessary to enlarge the radiators on this account, because the warm air, rising from below, is generally sufficient to offset any increase in heat transmission due to this cause. In case of very high buildings, however, where the upper floors are more exposed to the wind, it may be necessary to allow for the thinning of the walls and also to increase the factor for leakage.

When there is a low attic or roof space, in which steam pipes are carried, no additional radiation is required to offset the effect of a cold ceiling on the upper story.

In case of a roof space without steam pipes, increase the radiation about 15 per cent. When there is a solid roof, without air space, use the factor from Tables VI or VIII which corresponds most nearly to the actual conditions.

Example.—A room in an office building has a wall exposure of 450 square feet and a glass exposure of 100 square feet. The walls consist of 8 inches of brick, with sandstone facing, plastered on the inside. The windows have single sashes, and the exposure is east.

What will be the total heat loss per hour for a temperature difference of 70 degrees?

Factor for wall....	$22.5 \times 1.3 = 29.2$
Factor for window.....	= 84
Factor for leakage.....	= 1.2
Factor for exposure.....	= 1.1

from which the total heat loss is found to be

$$[(450 \times 29.2) + (100 \times 84)] \times 1.2 \times 1.1 = 28,433 \text{ B. T. U.}$$

93. *Class II.*—Buildings of this type are usually constructed of solid brick, or concrete, without insulating finish of any kind upon the inside walls, except in special rooms such as offices, drafting rooms, etc.

TABLE V.

HEAT LOSS THROUGH WALLS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 70 degrees.

Solid walls.	Heat loss in B. T. U.
8-inch brick	28
12 " "	22
16 " "	18
20 " "	15
24 " "	14

For reinforced concrete, multiply the factors in Table V. by 1.3.

For ordinary wooden construction with paper and clapboards on the outside and lath and plaster on the inside, use a factor of 21 for a temperature difference of 70 degrees.

For roof construction of different kinds, use Table VI.

TABLE VI.

HEAT LOSS THROUGH ROOFS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 70 degrees.

Roof construction.	Heat loss in B. T. U.
Slate on matched boards.....	21
Iron on matched boards.....	12
8-inch hollow tile, 1-inch concrete, tar and gravel	28
6-inch concrete, cinder fill, tar and gravel.....	38

For window surface, see Table II.

With wooden sashes the factor for leakage may be taken as 1.3, while for iron sashes, which are much tighter, it may be reduced to 1.1. In shops and factories where the floors are undivided by partitions, and the walls exposed to all four points of the compass, a factor of 1.16 may be used when correcting for exposure. For other temperature differences than 70 degrees use the factors given in Table III.

94. *Class III.*—The conditions here are much the same as in Class II., except the entire building is treated as a single room and a lower inside temperature is maintained.

TABLE VII.

HEAT LOSS THROUGH WALLS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 60 degrees.

Solid walls.	Heat loss in B. T. U.
4-inch brick	38
8 " "	24
12 " "	19
16 " "	16

For reinforced concrete multiply the factors in Table VII. by 1.3.

TABLE VIII.

HEAT LOSS THROUGH ROOFS.

Heat loss in B. T. U. per square foot of surface per hour for a temperature difference of 60 degrees.

Roof construction.	Heat loss in B. T. U.
Patent roof, (tar and gravel).....	18
6-inch hollow tile, 2-inch concrete, tar and gravel	22
4-inch concrete, cinder fill, tar and gravel.....	36
Unlined metal.....	78

For tile and concrete floors on earth use a transmission factor of 6 for ordinary conditions.

For single windows, and a temperature difference of 60 degrees, use a transmission factor of 72.

Leakage may be treated the same as in Class II., using the factors 1.3 and 1.1 for wooden and iron sash, respectively.

The factor for exposure may be taken as 1.16, considering the whole building as a single room.

95. *Class IV.*—Buildings of this type are confined chiefly to foundries, boiler shops, forge shops, etc., where the normal inside temperature may be comparatively low.

Tests show that the transmission losses through unlined corrugated metal are very nearly the same as for a single window, and for all practical purposes no distinction need be made between the walls, windows, or roof in buildings of this kind.

Taking the transmission factor as 60 for a temperature difference of 50 degrees, and multiplying by 1.3 and 1.16 for leakage and exposure, respectively, gives a total loss of $60 \times 1.3 \times 1.16 = 90.5$ B. T. U. per square foot of gross area per hour. To this should be added a transmission of about 5 B. T. U. per square foot of floor area.

In general, a heat loss of 100 B. T. U. per square foot of gross wall and roof area will safely cover all ordinary conditions where the outside temperature does not drop much below zero.

Radiation.

96. *Computing Direct Radiation.*—The square feet of direct radiation for a given room or building are found by dividing the total heat loss through transmission and leakage per hour, in B. T. U., by the efficiency of the type of radiator to be used. This simply allows for maintaining the normal temperature after the room is warmed. When the building is allowed to cool off at night, from 10 to 15 per cent additional radiation should be provided, for quick warming in the morning.

Experience has shown that but little if any more fuel is required for warming a building continuously than when it is allowed to cool off at night. Although no heat is taken from the boiler when the radiation is shut off, the building and its contents are constantly giving out stored heat which must be replaced by forcing the plant before the rooms can again be raised to their normal temperature.

For this reason, the heat in large buildings is usually kept on for a portion of the night, at least, to prevent the temperature from falling to any great extent.

In cases of this kind it is not common to allow any additional capacity for "warming up," especially as the radiating surface is proportioned for the coldest weather, and is therefore in excess of the requirements for most of the time.

Furthermore, the lower temperature of the building adds to the efficiency of the radiation, thus increasing the power of the plant still more under these conditions.

97. Efficiency of Direct Radiation.—The efficiency of a radiator depends upon the difference in temperature between the enclosed steam or water and the surrounding air, and also upon the velocity of the air passing over the heating surface. For the average conditions of direct heating the following rates of transmission may be used for different types of radiation. The variation is due to the form of surface, which affects the freedom of air flow, as noted above.

TABLE IX.

HEAT TRANSMISSION FROM DIRECT RADIATORS TO AIR.

Type of radiator.	B. T. U. given off per hour per square foot of surface per degree difference in temperature between the radiator and surrounding air.
Four-column radiators, 38 inches to 45 inches high.....	1.5
Three-column radiators, 38 inches to 45 inches high.....	1.6
Two-column radiators, 38 inches to 45 inches high.....	1.7
Three and four column radiators, 18 inches to 26 inches high.	1.7
Wall radiators.....	1.85
Wall coils, 1-inch to 2-inch pipe.....	1.95

Assuming the temperature of the surrounding air as 70 degrees, the following "working efficiencies" may be used for steam and water, as given in Tables X. and XI., respectively.

TABLE X.

EFFICIENCY OF DIRECT STEAM RADIATORS.

Type of direct radiating surface.	B. T. U. given off per square foot of surface per hour.	
	2 pounds gauge pressure.	Atmospheric pressure.
Two-column radiators, 38 inches high	255	240
Wall radiators.....	280	260
Pipe coils.....	290	275

TABLE XI.
EFFICIENCY OF DIRECT HOT-WATER RADIATORS.

Type of direct radiating surface.	B. T. U. given off per square foot of surface per hour for different <i>average</i> temperatures of water.		
	170 degrees.	180 degrees.	200 degrees.
Two-column radiators, 38 inches high	170	185	220
Wall radiators.....	185	200	240
Pipe coils.....	195	215	250

As a matter of fact, in computing the radiating surface for a system to be used in connection with a power plant, but little difference is made in the amount furnished, whether for low-pressure gravity, vacuum steam, or forced hot water, although the efficiencies for the usual conditions are given in the tables as 255, 240 and 220 B. T. U., respectively.

While it is common to carry from 2 to 5 pounds pressure in gravity plants using live steam it is desirable in exhaust heating to reduce this to a point as low as consistent with a proper discharge of air and condensation against atmospheric pressure. As air venting and steam circulation are somewhat more positive with a vacuum system the actual radiator efficiencies will be very nearly the same in the average plant.

Experience shows that forced hot water circulated at an average temperature of 200 degrees, with a drop not exceeding 20 degrees, (210 to 190 degrees), will have an actual efficiency approximating that of steam at atmospheric pressure, although the theoretical efficiency is somewhat less. For these reasons some engineers make a practice of using the same radiator efficiency for all three classes of work. When a live steam heater is provided, initial water temperatures considerably higher may be employed, if desired. In the earlier systems of forced hot-water heating, lower temperatures were carried, thus requiring larger radiators than for steam. Under the conditions of modern practice, efficiencies corresponding to those given for steam at atmospheric pressure in Table X. may generally be used for all three classes of direct heating in connection with power work.

When a cast-iron radiator is concealed by grille work, the surface should be increased at least 25 per cent and the openings so

arranged that the air will circulate freely over it. Coils hung near the ceiling should be increased in size from 30 to 40 per cent where it is desired to maintain a comfortable temperature near the floor.

98. Efficiency of Indirect Gravity Radiation.—If it is simply desired to warm a room by indirect radiation, without reference to ventilation, the direct radiating surface may be computed and the result multiplied by 1.5.

When definite conditions as to air temperature and volume are required, the following formulae may be used.

$$C = \frac{L \times 55}{T - t}$$

$$T = \frac{L \times 55}{C} + t$$

$$H = \frac{T \times C}{55}$$

in which

C = cubic feet of air required per hour, at temperature T , to bring in the necessary heat (L) to warm the room.

L = heat loss from the room, in B. T. U. per hour, due to transmission and leakage.

T = required temperature of air supply (C) to warm the room to 70 degrees.

H = heat required, in B. T. U. per hour, to raise the air supply (C) from outside temperature to that of admission (T).

t = room temperature, commonly taken as 70 degrees.

Example.—The total heat loss from a room by transmission and leakage in zero weather is 30,000 B. T. U. per hour. The cubic contents of the room is 12,000 feet, and it is desired to change the air four times per hour.

What volume of air must be supplied, and at what temperature must it be admitted to warm the room to 70 degrees?

What amount of indirect radiating surface will be required?

Applying the preceding formulæ to the present case we have,
 $C = 12,000 \times 4 = 48,000$ cubic feet per hour.

$$T = \frac{30,000 \times 55}{48,000} + 70 = 104 \text{ degrees.}$$

$$H = \frac{104 \times 48,000}{55} = 91,000 \text{ B. T. U. in round numbers.}$$

Having determined the value of H , the square feet of indirect radiation required to warm the room may be found by the equation,

$$R = \frac{H}{E}$$

in which

R = square feet of radiation.

E = efficiency of radiator employed.

The value of E depends upon the temperature of the radiator, the average temperature of the air passing over it, and the velocity of flow.

With a velocity of 270 feet per minute, about 2.1 B. T. U. will be given off by the radiator per square foot of surface per hour, per degree difference between the *average* temperature of the air and that of the radiator. The average air temperature in

the present case is $\frac{0+104}{2} = 52$ degrees. Therefore, if steam at atmospheric pressure is used (212 degrees), the efficiency E will be $(212-52) \times 2.1 = 336$ B. T. U.

For other velocities the efficiency will vary and may be computed by use of Table XII.

TABLE XII.

HEAT TRANSMISSION FROM INDIRECT RADIATORS TO AIR.

Velocity of air over heating surface, in feet per minute.	B. T. U. given off per hour per square foot of surface per degree difference in temperature between the radiator and surrounding air.
150	1.54
200	1.82
250	2.02
300	2.22
350	2.42
400	2.60
450	2.72
500	2.81

In practice it has been found safe in gravity heating to allow efficiencies of about 350 B. T. U. for dwellings, 450 B. T. U. for cottage hospitals, and 550 B. T. U. for schools, in the case of steam, with corresponding values of 260 B. T. U., 340 B. T. U., and 400 B. T. U. for hot water.

99. Efficiency of Induction Heaters.—Heaters of this type have already been described in connection with the warming and ventilation of bar rooms, laundries, loft buildings, etc., where discharge ventilation is produced by means of a fan and the air supply drawn in through heaters, as shown in Figs. 29 and 42, due to the suction effect of the exhaust.

The conditions here are much the same as already described for indirect radiation, except the velocity of the entering air is usually somewhat greater than in gravity work, which increases the efficiency slightly.

In the case of direct-indirect radiators, of the type shown in Fig. 15, an efficiency of 2 B. T. U. per square foot of surface per hour per degree difference between the radiator and average air temperature may be assumed, while with pin radiation and similar patterns, which break up the air currents more thoroughly, an efficiency of 2.4 B. T. U. may be used.

Example.—A bar room $20 \times 40 \times 15$ feet has a heat loss, due to transmission and leakage of 20,000 B. T. U. per hour in zero weather. Exhaust ventilation is to provide eight changes of air per hour.

What will be the required heating surface, in the form of direct-indirect radiators? Applying the formulae as before, we have,

$$C = 20 \times 40 \times 15 \times 8 = 96,000 \text{ cubic feet of air per hour.}$$

$$T = \frac{20,000 \times 55}{96,000} + 70 = 82 \text{ degrees.}$$

$$H = \frac{82 \times 96,000}{55} = 143,000 \text{ B. T. U.}$$

$$\text{Average air temperature, } \frac{0 + 82}{2} = 41 \text{ degrees.}$$

Temperature of steam at atmospheric pressure, 212 degrees.

$$E = (212 - 41) \times 2 = 342 \text{ B. T. U.}$$

$$R = \frac{143,000}{342} = 418 \text{ square feet of radiation.}$$

In the case of laundries and loft buildings the process is practically the same, except that direct radiation is commonly employed for heating and the induction stacks used simply for ventilating purposes. In this case, the required heat is determined directly by the formula $H = \frac{T \times C}{55}$ in which T is commonly taken as 72 to 74 degrees, unless a portion of the warming is also done by the induction heaters.

When ventilation, only, is required, an efficiency of (212—36) $\times 2.4 = 422$ B. T. U. per square foot of surface per hour may be allowed for pin radiation with steam at atmospheric pressure.

Steam Piping.

100. Systems Employed.—Two general systems of steam piping are employed in large buildings, known as the “up-feed” and “down-feed” systems. In the first, shown in diagram in Fig. 61, the distributing mains are run at the basement ceiling and the supply risers carried to the top of the building. In the gravity system either the single or double radiator connection may be used, as indicated in the cut. While this system is extensively used, under certain conditions, it is not so well adapted to tall buildings as the down-feed, shown in Fig. 62. In this case a single riser of sufficient size to supply the entire system, is carried to the top of the building, where it branches in the roof space, or at the ceiling of the upper floor, and connects with the various supply drops or “risers.” This has the advantage of removing a considerable amount of piping from the basement ceiling, which is usually over-crowded with ventilating ducts and other equipment, and also serves to supply a sufficient quantity of heat just below the roof to offset the effect of a cold ceiling on the upper floor. The down-feed system is especially adapted to the single-pipe radiator connection, because the flow of steam and condensation being in the same direction, much smaller risers may be used. The up-feed, with single-pipe radiators, is limited to buildings three or four stories in height, owing to the excessive size of the supply risers passing through the lower stories. This, however, is avoided to a large extent when the two-pipe system

is employed, but this arrangement adds another set of risers for bringing back the condensation.

A scheme often employed in office buildings is shown in Fig. 63, in which case the lower floor is supplied by a separate up-feed system and the remainder of the building by a down-feed system.

The advantage of this is the elimination of supply risers from the banking rooms and offices on the first floor, while the drips from the floors above are so small that they may be easily concealed.

Sometimes they are grouped together, thus reducing the number and assisting still more in the process of elimination.

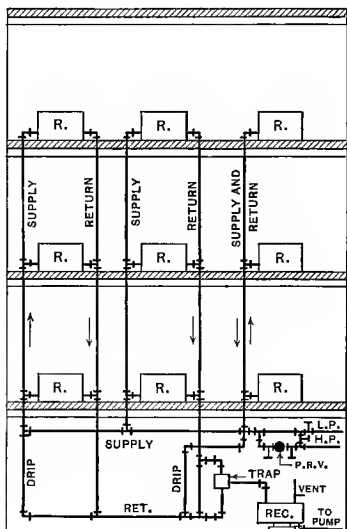


Fig. 61.

Fig. 61. Diagram of Piping for Upward Supply.

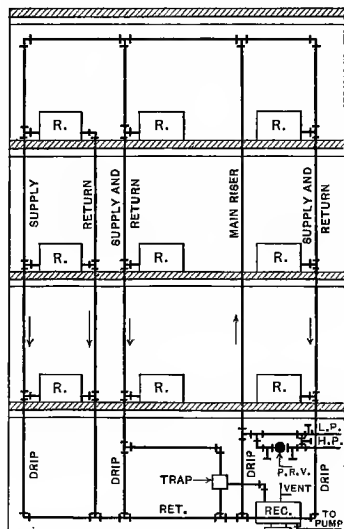


Fig. 62.

Fig. 62. Diagram of Piping for Downward Supply.

101. Expansion of Risers.—An important matter to be considered in this class of work is the expansion of the risers.

This may be provided for either by loops and swivels or by slip joints placed six or eight stories apart. Expansion loops require less care when once installed, but are rather difficult to conceal in buildings of modern fire-proof construction. Slip

joints, on the other hand, are easily installed and incased, but are more likely to bind or leak. However, with the low pressures carried in work of this kind the slip joint seems to give good satisfaction and is quite generally used in the best class of work.

102. Pipe Sizes for Gravity Return Systems.—Pipe sizes for steam systems in which the condensation returns to the receiving tank by gravity, are usually based upon the square feet of radia-

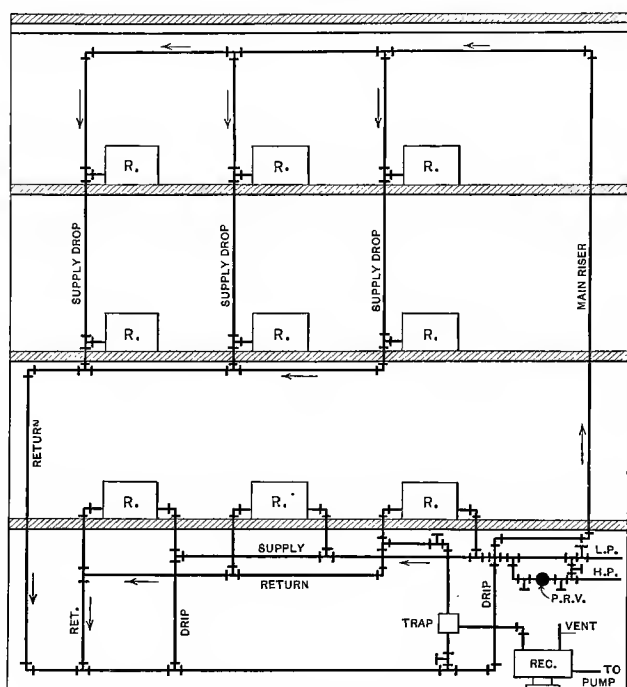


Fig. 63. Diagram of Piping for Combined Upward and Downward Supply.

tion which they supply, the length of run, and the drop in pressure at the extreme end of the line.

Table XIII. gives sizes of steam and return pipes for varying amounts of direct radiation, based upon drops in pressure of one-quarter pound in 100, 200 and 400 feet length of run. Knowing the pressure to be carried and the distance to the farthest radiator, the allowable drop in pressure is assumed and the main

or branch proportioned accordingly. Ordinarily the drop in pressure should not exceed one pound at the extreme end of the line, with a gravity return to the receiver. As the condensation in systems of this kind is usually trapped into a vented receiving tank, against atmosphere, there should be an excess of pressure in the return sufficient to produce a free discharge of the trap into the receiver. Furthermore, a variation of pressure exceeding one pound in different parts of the system is difficult to balance by the use of siphons and sealed returns.

Under ordinary conditions an initial pressure of two pounds gauge may be carried on the heating system without materially increasing the back pressure on the engines over that when exhausting to atmosphere. This allows a drop in pressure of one pound at the end of the line and furnishes an excess of one pound above the atmosphere for the discharge of air and condensation. In many plants, however, the piping is proportioned for a working pressure of one pound or less. It may be noted in this connection that a plant designed for a maximum pressure of two pounds in zero weather will operate under a considerably lower pressure for a large proportion of the heating season. With a return-line vacuum system greater drops in pressure may be employed for reasons described later.

TABLE XIII.

PIPE SIZES FOR DIRECT STEAM HEATING.

Square feet of direct radiation.			Size of pipe.		
Drop in pressure ¼-pound in 100 feet.	Drop in pressure ¼-pound in 200 feet.	Drop in pressure ¼-pound in 400 feet.	Steam supply, inches.	Dry return, inches.	Wet return, inches.
60 to 80	40 to 60	20 to 40	1	1	¾
80 " 150	60 " 100	40 " 70	1½	1	1
150 " 250	100 " 160	70 " 120	1½	1½	1
250 " 500	160 " 350	120 " 250	2	1½	1½
500 " 1,000	350 " 700	250 " 500	2½	2	1½
1,000 " 1,600	700 " 1,100	500 " 800	3	2½	2
1,600 " 2,300	1,100 " 1,600	800 " 1,200	3½	2½	2
2,300 " 3,300	1,600 " 2,300	1,200 " 1,700	4	3	2½
3,300 " 6,000	2,300 " 4,000	1,700 " 3,000	5	3	2½
6,000 " 10,000	4,000 " 7,000	3,000 " 5,000	6	3½	3
10,000 " 14,000	7,000 " 10,000	5,000 " 7,000	7	3½	3
14,000 " 20,000	10,000 " 14,000	7,000 " 10,000	8	4	3

103. *Sizes of Risers.*—When the down-feed system is used, with single-pipe radiators, the supply risers may be taken from

Table XIII., using one-quarter pound drop in 100 feet for buildings up to ten stories in height, and one-quarter pound drop in 200 feet above that. Risers of this kind are reduced in size as they pass downward, according to the amount of radiation supplied, but not in exact proportion. As they carry both steam and condensation it is best to make the lower half one size larger than called for by the table, in order to reduce the velocity and prevent the water from being drawn into the radiators on the lower floors. In buildings over ten stories in height, two-thirds of the riser may be enlarged in this manner.

The drips at the bottoms of the risers are commonly made somewhat smaller than the horizontal returns, due to their vertical position, and the following proportions may be used under ordinary conditions.

TABLE XIV.

SIZES OF DRIPS.

Size of supply riser at top, inches.	Size of drip pipe, inches.
1 to $1\frac{1}{2}$	$\frac{3}{4}$
2 " $2\frac{1}{2}$	1
3 " $3\frac{1}{2}$	$1\frac{1}{4}$
4 " 5	$1\frac{1}{2}$

In the up-feed system, with two-pipe radiator connections, the supply risers may be taken from Table XIII., as they carry no condensation except that which is formed in the pipes themselves.

In case of the single-pipe connection, the steam and water are flowing in opposite directions, and the risers must be made much larger in order to reduce the velocity and prevent interference with the drainage. For this class of work Table XV. may be used.

TABLE XV.

SIZES OF UP-FEED RISERS.

Square feet of direct radiation.	Size of riser, inches.
20 to 40	1
40 " 80	$1\frac{1}{4}$
80 " 100	$1\frac{1}{2}$
100 " 150	2
150 " 250	$2\frac{1}{2}$
250 " 350	3
350 " 500	$3\frac{1}{2}$

104. *Pipe Sizes for Indirect Stacks.*—The tables of pipe sizes already given may be used with sufficient accuracy for indirect gravity heating stacks, by counting each square foot of indirect surface as two of direct.

Vacuum Systems.

105. *Types.*—These are of two general types, known as “air-line” and “return-line” systems. In the first, shown in diagram in Fig. 64, the steam and return connections may be the same as with a gravity system, and either the single or double radiator connection used as preferred. The suction, or vacuum, in this

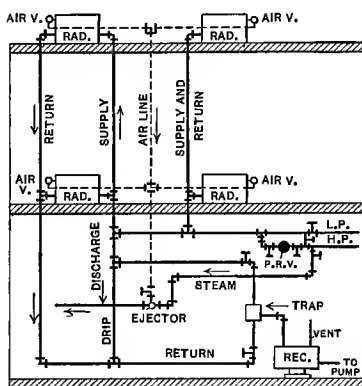


Fig. 64.

Fig. 64. Diagram of Air-Line Vacuum System.

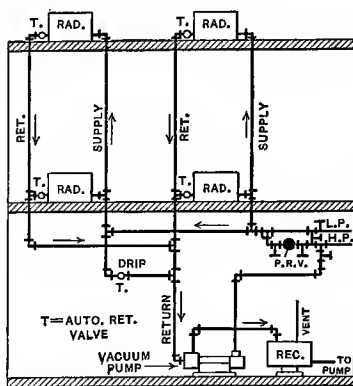


Fig. 65.

Fig. 65. Diagram of Return-Line Vacuum System.

case, is applied to the air valve, and does not affect either the pressure within the radiator or the return of the condensation.

With the “return-line” system shown in Fig. 65, it is necessary to use the double or two-pipe radiator connection. In this case the usual return valve is replaced by an automatic valve, which allows the passage of air and water, but closes in the presence of steam.

A vacuum pump is attached to the main return pipe, drawing out both air and water, which are discharged into a separating or receiving tank. From here the air passes to the atmosphere

and the water is returned to the boilers in the usual manner by a separate feed pump.

106. Advantages and Limitations.—Vacuum systems of both types have special fields in which they may be operated to advantage, but this does not mean their use is advisable under all conditions.

The air-line system is adapted to compact plants like office buildings where the ground area is small compared with the cubic contents. When used in connection with exhaust steam heating, where the condensation is trapped to a vented receiver, the principal advantage is a quick and positive removal of air from the system, as sufficient steam pressure must be carried in the radiators to discharge the condensation into the receiving tank against atmospheric pressure.

Under these conditions, however, the ordinary automatic air valve, with drip-line connection, will accomplish the same result, although not so rapidly, perhaps, at low pressures.

By using an independent receiver and return pump in place of the water-line trap, and running on a closed system, it is possible to carry pressures below that of the atmosphere, thus giving a slight range of temperature control and avoiding additional back pressure on the engines.

When a system for gravity return is well designed and provided with pipes of ample size, there is very little to be gained by the use of an air-line system, especially in connection with automatic temperature regulation.

The return-line system, under certain conditions, has several decided advantages over gravity heating. It is especially adapted to plants covering large ground areas and containing long runs of horizontal pipe. Connecting a suction to the main return reduces the required pitch and prevents water-hammer, which is likely to occur in plants of this kind operated on a gravity return.

Steam may be circulated at or slightly below atmosphere, thus relieving instead of adding to the back pressure.

A plant equipped with this system may be warmed up quickly in the morning, and when the radiators are provided with graduated valves at the steam inlets, is capable of a fair degree of hand regulation.

It will be seen by an inspection of Figs. 64 and 65 that the

vacuum only exists between the pump or exhauster and the automatic valves at the radiators. The flow of steam through the supply piping is due simply to the difference in pressure caused by the condensation of steam in the radiators, the same as in the gravity system. Lower pressures are possible in this case because it is not necessary for the air or condensation to discharge against the atmosphere, thus making it possible to lower the entire working range.

107. Vacuum Carried.—Only enough vacuum should be carried to ensure a prompt removal of air and water from the radiators and to lower the initial steam pressure in the heating system to approximately that of the atmosphere. A high vacuum tends to produce a leakage of steam through the automatic valve into the return, and also to cause a part of the condensation to flash into steam when the difference in pressure

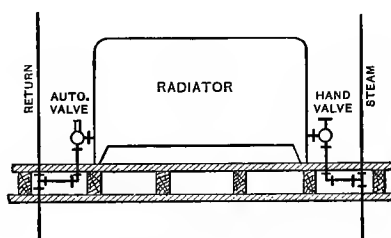


Fig. 66. Radiator Connections for Return-Line Vacuum System.

between the radiator and return pipe is too great. Unless there is an abundance of exhaust for heating purposes, the presence of steam in the main return is a constant source of waste, due to the necessity of condensing it by means of a water jet.

Ordinarily a vacuum of about 2 inches is carried at the most distant radiator. This may require a vacuum ranging from 3 or 4 inches to 20 inches at the pump depending upon the lift and length of run. Under average conditions, however, this will not exceed 8 or 10 inches, although sufficient pump capacity is usually furnished for a 20-inch vacuum.

108. Special Equipment.—In case of the air-line system the vacuum is produced either by a steam jet or electrically driven vacuum pump, depending upon the size of the plant.

The air valves used upon the radiators are of special design and form part of the regular equipment.

In the return-line system the air is removed through the automatic return valve, together with the condensation, as already stated. Typical connections for the radiators and coils are shown in Figs. 66 and 67, the latter illustrating a common method of catching dirt and scale before it enters the valve. The return from each radiator is provided with an automatic valve; also the drips from the bottoms of all supply risers and the ends of all steam mains. Pump connections are shown in Fig. 68. This equipment includes a strainer in the main return, with cold water jet, a controller for regulating the speed of the pump to

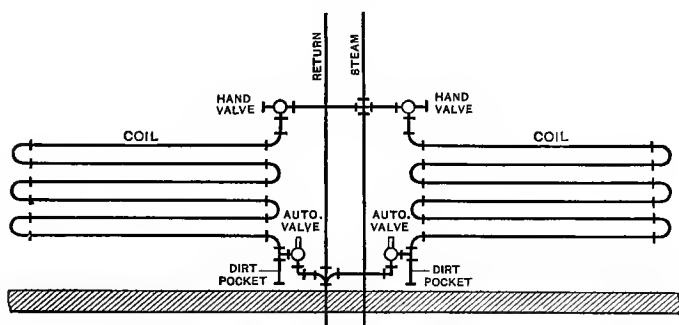


Fig. 67. Coil Connections for Return-Line Vacuum System.

maintain a constant vacuum of the desired strength, and gauges for indicating the supply and return pressures in the heating system. The usual vented receiver commonly takes the place of a special separating tank, although the latter is required when the condensation is discharged into an open feed-water heater.

Sometimes the arrangement of a plant is such that the radiation occurs in two or more widely separated groups, and the returns from different buildings are connected with a single vacuum pump in the power house. It is evident in cases of this kind that the buildings nearer the pump will have a considerably higher vacuum unless means are taken to equalize the pressures in the different return lines. This may be done by the arrangement shown in Fig. 69, which reduces the vacuum

in the nearest group and traps the condensation from it into the high vacuum line from the more distant one.

109. *Pipe Lines and Sizes.*—As already stated, the method of running the supply and return piping does not differ from the

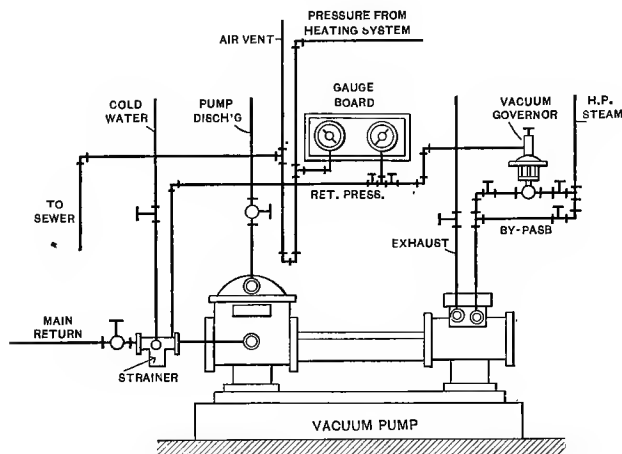


Fig. 68. Detail of Connections at Vacuum Pump.

ordinary gravity system, except the radiators must always have a double connection when a return-line vacuum is employed.

One advantage of the latter method is the ability to lift the

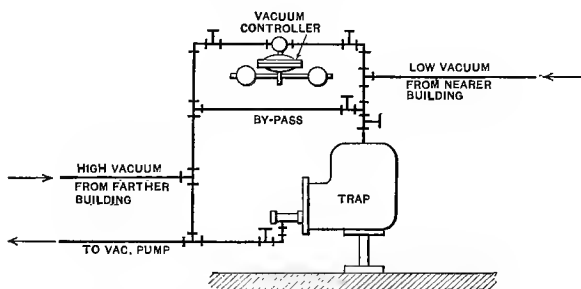


Fig. 69. Detail of Connections for Vacuum Controller.

condensation a certain height, thus making it possible to carry the return overhead as shown in Fig. 70, which is often a matter of much convenience.

While somewhat smaller pipes are often used in vacuum heating than with a gravity return, it is better to make the reduction chiefly in connection with the return pipes. Table XIII. may be used for the supplies the same as in gravity heating, although a larger drop in pressure may be allowed owing to the fact that the air and condensation are discharged below atmosphere, and

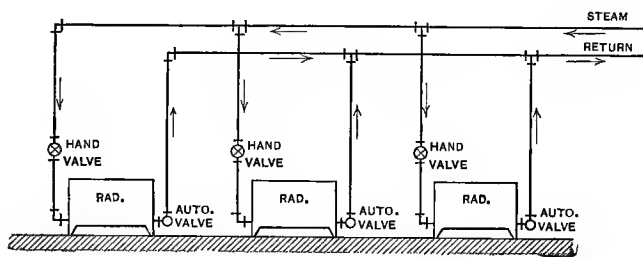


Fig. 70. Vacuum System with Overhead Return.

pressures in the return system are equalized. Table XVI. gives sizes of return pipes for different amounts of direct radiation for return lines from 200 to 600 feet in length.

TABLE XVI.

RETURNS FOR VACUUM HEATING.

Square feet of direct radiation.			Size of return, inches.
200-foot run.	400-foot run.	600-foot run.	
250	200	160	$\frac{3}{4}$
800	600	500	1
2,000	1,500	1,200	$1\frac{1}{4}$
4,500	3,500	3,000	$1\frac{1}{2}$
9,000	7,000	6,000	2
15,000	12,000	10,000	$2\frac{1}{2}$
25,000	20,000	16,000	3
38,000	32,000	26,000	$3\frac{1}{2}$

The sizes in the above table allow for a small amount of condensation in the returns themselves, and are somewhat larger than given by some manufacturers of vacuum equipment.

110. *Vacuum Pumps.*—Both direct-acting steam pumps of the single cylinder type, and motor or turbine driven rotary pumps are used in connection with vacuum heating.

While the former is wasteful in the use of steam it is easily

regulated to maintain any desired vacuum and is extensively used for this purpose. When there is an abundance of exhaust from the main engines for all heating purposes it is usually more economical to employ a motor-driven pump in plants of large size.

Table XVII. gives the required size of vacuum pump for different amounts of direct radiation. In this table account has been taken of the probable number of automatic valves as well as the amount of radiating surface.

Different makes of pumps will vary somewhat in their dimensions for the same capacity, but the following, rated on a steam pressure of 100 pounds, will serve as a guide in making a selection.

TABLE XVII.

VACUUM PUMPS.

Square feet of direct radiation.	Size of pump.			Size of openings.			
	Diameter steam cylinder, inch.	Diameter water cylinder, inch.	Length of stroke, inch.	Steam, inch.	Exhaust, inch.	Suction, inch.	Discharge, inch.
2,000	3	4	4	$\frac{1}{2}$	$\frac{3}{4}$	2	$1\frac{1}{2}$
4,000	4	5	5	$\frac{1}{2}$	$\frac{3}{4}$	3	$2\frac{1}{2}$
6,000	4	6	5	$\frac{1}{2}$	$\frac{3}{4}$	3	$2\frac{1}{2}$
8,000	5	6	6	$\frac{1}{2}$	$\frac{3}{4}$	3	$2\frac{1}{2}$
10,000	$5\frac{1}{2}$	6	6	$\frac{1}{2}$	1	$3\frac{1}{2}$	3
12,000	6	7	7	$\frac{1}{2}$	1	$3\frac{1}{2}$	3
15,000	$5\frac{1}{2}$	7	8	$\frac{1}{2}$	1	4	$3\frac{1}{2}$
20,000	6	8	8	$\frac{1}{2}$	1	4	$3\frac{1}{2}$
25,000	7	8	10	$\frac{1}{2}$	1	4	$3\frac{1}{2}$
30,000	7	9	10	$\frac{1}{2}$	1	6	$3\frac{1}{2}$
35,000	8	9	12	1	$1\frac{1}{2}$	6	5
40,000	8	10	12	1	$1\frac{1}{2}$	6	5

When the pump suction is larger than the corresponding return it should be bushed down to the same size. Vacuum pumps like all other special equipment of a similar kind should always be furnished in duplicate in order to guard against accident.

Forced Hot-Water Heating.

111. *General Advantages.*—Hot-water heating under gravity conditions is not adapted to large buildings owing to the weakness of the force which produces a circulation through the system.

This difficulty, however, is easily overcome by placing a pump in the circuit, by means of which the water may be forced through the mains and branches at any velocity desired.

The principal advantage of a heating system of this kind over steam is the wide temperature range of the heating medium, thus making it possible for the engineer to gauge the capacity of his plant to varying outside weather conditions. This results not only in greater comfort to the occupants of the building, but also reduces the waste of heat through over-heating and open windows. With automatic temperature control there is probably very little choice between steam and hot water, as regards comfort and cost of operation; but with hand control, even in the case of vacuum systems equipped with graduated valves, hot water has the advantage in these respects in the average building.

There may be other factors, however, which are decidedly in favor of steam, so that each plant should be studied from all points before making a decision between the two systems.

The objection formerly raised regarding increased radiating surface is now overcome by circulating the water at practically the same temperature as low-pressure steam in the coldest weather.

112. Systems of Piping.—Typical arrangements for forced hot-water heating are shown in diagram in Figs. 71 and 72. The important points to be considered in laying out a system of piping are positive and rapid circulation through the radiators; a uniform resistance through all parts of the system, avoiding so far as possible the necessity of throttle valves to prevent short-circuiting; and in general with other systems of piping, economy of installation.

One common arrangement is to carry a circuit main around the basement, connecting both supply and return risers into the same pipe and depending upon the difference in pressure between the points of connection to produce a flow through the radiators. The objection to this is the cooling effect of the return water as the distance from the heater increases, thus requiring larger radiators to offset it.

The plan illustrated in Fig. 71 operates upon this general principle, but is so arranged as to reduce the cooling effect to a

point, where with a rapid circulation, an increase of radiation is not necessary.

In this case a main supply riser is carried directly to the top of the building and provided with an air trap. From here the pipe passes downward, with a branch taken off at each story, connecting with a corresponding return drop at the other side of the building. The radiators are connected up for supply and return as indicated, and, as the number on each branch is comparatively small, the cooling effect is not appreciable. In very

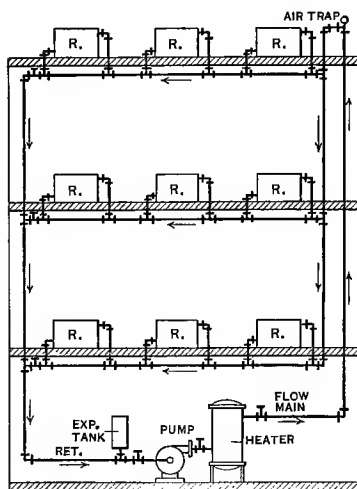


Fig. 71.

Fig. 71. Hot-Water System with Shunt Supply.

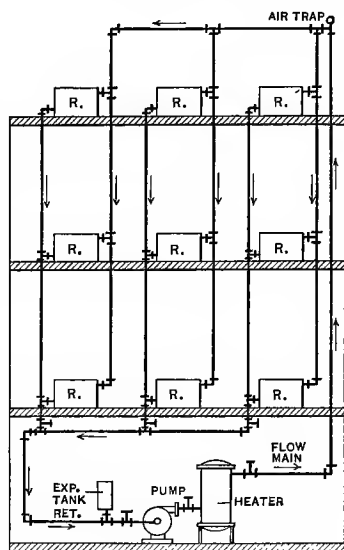


Fig. 72.

Fig. 72. Hot-Water System with Downward Supply.

large buildings, however, it may be necessary to divide the radiation on each floor into two or more groups in order to keep up the temperature at the end of the line. While a separate riser to the top of the building with a downward feed is more expensive to install, it has the advantage of preventing air in any quantity from reaching the radiators, and also makes the length of the circuit, from pump discharge to suction inlet, through each series of radiators the same, thus equalizing the

resistance and tending to produce an even circulation through all parts of the system. Finer adjustment may be made, if required, by the use of throttle valves placed in the branch from each floor near its connection with the main return. Sometimes the returns are carried to the basement separately, in order to make these valves accessible to the engineer at all times. The horizontal branches between the supply and return drops may be carried either behind removable bases or in a space beneath the finished floor. In the two-pipe arrangement, shown in Fig. 72, the main riser and air trap are the same as in the system just described, but the discharge from each radiator passes directly into the return without entering any of the others. This avoids the cooling effect previously noted, and also makes it possible to force the water through the radiators at a higher velocity in case it is desired to increase their capacity temporarily.

Here, as in Fig. 71, the arrangement is such that the length of the circuit through each riser is practically the same.

113. Pipe Sizes.—The first step in computing the size of main for a given system is to determine its effective length, which is equivalent to the length of the circuit, from discharge to inlet of pump, plus the frictional resistance of bends, expressed in feet of additional pipe. In estimating the latter, short-pattern fittings may be taken as 9 and 17 feet for ells and tees respectively; while for long-turn fittings, these figures may be reduced to 4 and 9 feet. Swing checks may be taken from 25 to 50 feet according to size.

The actual length of the circuit, in Fig. 71, is from the pump discharge to the top of the main riser, across the building, down the main return drop, and back to the pump. With the arrangement of piping shown, this distance is the same through each group of radiators. In case of Fig. 72 the method of measurement is practically the same.

The next step is to compute the gallons of water to be circulated through the system per minute. This depends upon the B. T. U. to be supplied and the assumed drop in temperature, and is given by the formula

$$G = \frac{H}{8.3 \times T}$$

in which

G = gallons of water to be circulated per minute.

H = B. T. U. to be supplied per *minute*.

T = drop in temperature, commonly taken as 20 degrees under average conditions, although greater drops are sometimes employed.

The total B. T. U. to be supplied may be computed by the methods described or determined by multiplying the total radiating surface by its efficiency under the assumed conditions, increasing the result 20 per cent for pipe losses. In each case the heat quantity is to be reduced to B. T. U. per *minute*.

Velocities ranging from 5 to 10 feet per second are commonly employed in the mains, depending upon their size, with friction heads limited to 30 or 35 feet, although resistances somewhat higher may be operated against by using pumps designed especially for the work.

In case of the smaller branches the velocity should be less than in the mains in order to maintain the same drop in pressure per unit of length. Velocities of 3 or 4 feet per second are common in pipes of small size. Smaller mains are less expensive to install, but increase the friction head and cost of pumpage; hence a compromise must be made to secure the most economical results.

As the static pressure of the water is the same on both sides of the pump the only work required is that of overcoming the frictional resistance within the mains. This pressure, however, must be considered in connection with the strength of the radiation and tightness of joints. In general, the static head should not exceed about 100 feet, which, with a differential pressure of 35 feet, makes a total pressure of $(100+35) \times 0.4 = 54$ pounds per square inch on the discharge side of the pump. In the case of tall structures, where it is necessary to limit the static pressure, the building is divided into sections of eight or ten stories, and each provided with its own equipment of pumps and heaters.

Table XVIII. gives the capacities, velocities, and friction heads for pipes of different size within the usual working range.

For conditions not included in the table, computations may be made with sufficient accuracy by means of the empirical formula

$$H = \frac{1.12(4v^2 + 5v)L}{1,200d}$$

in which

H = friction head, in feet.

v = velocity of flow, in feet per second.

L = length of pipe, in feet.

d = diameter of pipe, in inches.

TABLE XVIII.

FRICION HEADS FOR 100 FEET LENGTH OF PIPE.

	Diameter of pipe.	¾-inch.	1-inch.	1¼-inch.	1½-inch.	2-inch.	2-½ inch.
Gallons discharged per minute.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.	Velocity, in feet per second. Friction head, in feet.
2	1.4 .2	2.0 4.0	.	.	.	.	.
3	.2	.	.	.	.	.	.
4	.	.	1.6 2.0	1.8 2.6	.	.	.
5	.	.	2.4 3.6	.	.	.	.
6	.	.	.	.	.	.	.
8	.	.	.	2.1 3.1	1.6 3.5	.	.
10	.	.	.	.	.	.	.
12	.	.	.	.	.	.	.
14	.	.	.	.	2.2 3.9	.	.
16	.	.	.	.	2.8 5.0	.	.
18	.	.	.	.	.	2.0 3.1	.
20	.	.	.	.	.	4.1	1.2 2.4 4.0
30	.	.	.	.	.	.	3.3 4.6
40	.	.	.	.	.	.	2.1 2.9
50	.	.	.	.	.	.	3.9
60	.	.	.	.	.	.	2.7
70	.	.	.	.	.	.	3.9

[illegible]

For small differences, values may be interpolated in the table as illustrated in the following example.

Example.—A 10-story building is to be equipped with a hot-water system of the type shown in Fig. 71. What size of main will be required, and what should be the size of offtake to each floor under the following conditions?

Height of vertical sections of main, 140 feet.

Length of horizontal sections of main, 100 feet.

Square feet of radiation on each floor, 800 feet.

Average temperature of water, 200 degrees, with drop of 20 degrees in circuit.

Taking up the different steps in the order described we have—

Total length of circuit, $140+140+100+100=480$ feet, which may be taken as 500 feet in round numbers, to include the effect of fittings.

Quantity of water to be circulated =

$$\frac{800 \times 10 \times 220}{8.3 \times 20 \times 60} = 177 \text{ gallons per minute.}$$

Interpolating in Table XVIII., between the values for 150 and 200 gallons per minute, the approximate values for velocity and friction head for a 4-inch pipe are found to be 4.5 and 2.5 feet respectively, making a total head of $5 \times 2.5 = 12.5$ feet for the entire circuit, against which the pump must operate.

Each supply branch across the building must carry $\frac{177}{10} = 17.7$ gallons per minute under approximately the same friction head per unit of length.

Table XVIII. shows that a $1\frac{1}{2}$ -inch pipe will discharge 18 gallons per minute with a head of 3.5 feet, or a 2-inch pipe will discharge 20 gallons with a head of 1.2 feet. As the frictional resistance of the entire circuit is low, it would probably be best to use the smaller size and carry a differential pressure on the pump of $5 \times 3.5 = 17.5$ feet, which comes well within the working limit for efficient operation.

114. Expansion Tank.—This should have a capacity equal to at least twice the increase in volume of the water in rising

through 150 degrees in temperature and is given approximately by the equation

$$V = 0.025 \times R$$

in which

V = capacity of tank, in gallons.

R = square feet of radiation.

This takes into account the water in the piping and allows ample air-space in the tank.

As it is desired to carry temperatures at or above the boiling point, closed tanks are used, generally located near the pumps.

A typical arrangement for a tank of this kind is shown in Fig. 73, in which a constant air pressure is maintained above the water by a combination of relief valve and automatic air pump.

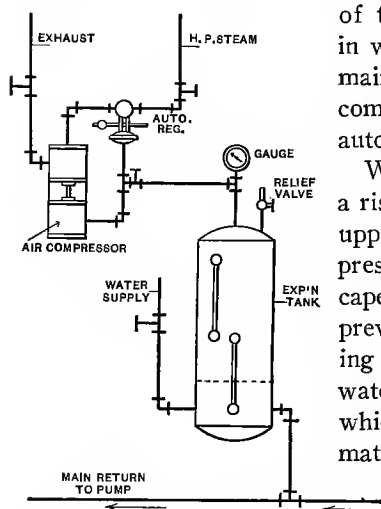


Fig. 73. Piping Diagram for Closed Expansion Tank.

When the water expands, due to a rise in temperature, the air in the upper part of the tank is compressed and a sufficient amount escapes through the relief valve to prevent the pressure from exceeding a given point. A fall in the water level reduces the air pressure, which is at once made up automatically by the compressor. Additional water is admitted from time to time, as needed, by means of city pressure or a small pump under hand control.

When for any reason it is desired to place the tank at the top of the system, the arrangement shown in Fig. 74 may be used. In this case air is admitted through a check valve opening inward under atmospheric pressure, and the water supplied as needed, by means of a float valve. In the Schott system shown in Fig. 75, an increase in pressure, due to expansion, opens a relief valve and allows the surplus water to flow into an open tank. When the pressure falls, water is forced back

into the system automatically by the balanced-column pump, shown near the relief valve.

The pressure carried on any system will depend upon the maximum temperature it is desired to maintain in the radiators.

115. Air Venting.—All high points in the piping system should be provided with automatic air traps, and key air valves should be furnished on all radiators. Any good form of float steam trap may be used for air venting, provided the discharge valve is exposed when placed bottom upward.

116. Heaters.—Live-steam heaters are preferable to hot-water boilers for two principal reasons; first, the temperature of the water is under better control, and second, because high pressure steam boilers are available either for power or heating purposes. Hot-water boilers, on the other hand, should be in duplicate to guard against accident, and must in any case be laid

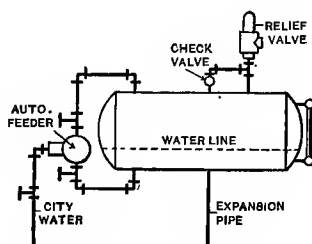


Fig. 74. Piping Diagram for Overhead Expansion Tank.

off during the non-heating season, thus adding to the cost of the equipment over that when steam alone is used.

Combined power and heating plants should, in general, be provided with both exhaust and live-steam heaters, the latter being of sufficient capacity to do the entire work, when the engines or turbines are not in use. When there is an abundance of exhaust for all heating purposes, there is no necessity for a live-steam heater unless it is desired to obtain a higher water temperature than is possible with steam at exhaust pressure. When the demands for heating exceed the supply of exhaust steam, for short periods, the deficiency may be made up by admitting live steam to the heater through a reducing valve.

If live steam is to be used in any considerable amount it is more economical to employ an independent heater and return the condensation to the boilers by gravity.

If trapped to the hot-well or a vented receiver, part of the water will break into steam upon release from the higher pressure and so pass off through the vapor pipe as waste heat.

Sometimes, when live steam is required only in extreme weather, the condensation is trapped into the exhaust heater where it is re-evaporated at the lower pressure.

If reciprocating engines are used, the exhaust should be passed

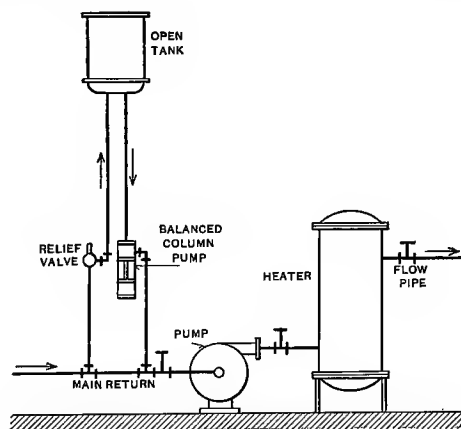


Fig. 75. Piping Diagram for Schott System.

through an efficient oil separator before entering the heater in order that the condensation may be returned to the boilers. This, however, is not necessary in the case of turbines, which require no oil in the steam passages.

If a pressure above that of the atmosphere is always maintained in the exhaust heater, and the relative elevations are right, the condensation may be trapped to the hot-well or receiving tank, otherwise a pump should be employed which makes drainage possible at all times without regard to the pressure within the heater. The exhaust from this pump, as well as from all other auxiliaries, should be utilized either in the main heater or in a special feed-water heater.

The tube surface of a heater is much more efficient when the water passes through the tubes with the steam on the outside, owing to its higher velocity over the heating surfaces.

The transmission of heat is effected not only by the velocities of the steam and water but also by the accumulation of grease upon the tubes, the dripping of condensation from one tube to another and also by the presence of air. While comparatively high efficiencies are obtained in modern condenser practice the same results are not possible in heaters of this type owing to the different conditions under which they operate. For the average velocities employed, and without special provision for the removal of air, except the usual vents, a transmission of 250 to 300 B. T. U. per square foot of surface per hour per degree difference between the temperature of the steam and the *average* temperature of the water may be assumed for brass tubes. With higher velocities of flow, clean tubes, and more efficient air venting, a transmission of 300 to 350 B. T. U. may be obtained. Examinations of a considerable number of plants recently installed, showed the heaters to be designed for transmission factors ranging from 250 to 300 B. T. U.

Various reported tests give considerably higher rates of transmission than are commonly obtained in practical work.

In order to secure the highest efficiency the heater should be so designed that the water will pass through the tubes under approximately the same velocity and pressure head as in the mains connecting with it.

117. Circulating Pumps.—The centrifugal or turbine pump driven by a direct-connected motor or steam turbine is best adapted to this work. In power plants generating their own electricity the motor drive is usually more economical and is commonly employed for day work. This should be supplemented by a steam turbine unit for night use or at other times when electric current is not available.

Each pump should be capable of doing the entire work in case of accident or repairs. The horse power required for driving the pump is given by the formula,

$$\text{H. P.} = \frac{F \times G \times 8.3}{33,000 \times E}$$

in which

F = total friction head pumped against, in feet.

G = gallons of water discharged per minute.

E = efficiency of pump.

118. *Pump and Heater Connections.*—A typical pump and heater equipment, showing the various connections, is illustrated in Fig. 76. Pump No. 1 is motor driven, while No. 2 is connected with a steam turbine for independent operation. Connections with the main return are such that they may be run separately or in series as desired.

Although each pump should be capable of moving the full volume of water under the required head, it is sometimes desir-

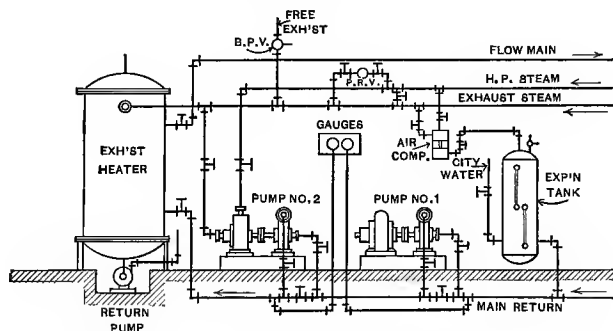


Fig. 76. Diagram of Pump and Heater Connections for a Single Low-Pressure Heater.

able to increase the radiator and heater efficiencies temporarily by circulating the water at a higher velocity. When the pumps are in series this may be done, as the pressure heads are added when operated in this way. The plant shown is provided with an exhaust heater, only; more heating capacity being furnished by admitting live steam through a reducing valve. The condensation is returned to the hot-well by a separate turbine-driven pump.

The heater connections for a plant using both exhaust and live steam are shown in Fig. 77, the high-pressure heater being arranged for a gravity return of condensation to the boilers. When the height from the water line to the bottom of the heater is

less than 10 feet, a live-steam connection should be made with an injector tee in the return, as shown at A, to overcome the drop in pressure in the heater under heavy service.

Variations in water temperature are secured by throttling the steam supply to the heater which reduces the pressure and allows the condensation to collect and partially submerge the tubes, thus reducing their efficiency. The two heaters are connected in series, and so valved that they may be operated separately or together.

119. "*Vacuo*" Hot-Water Heating.—This term is applied to systems where the exhaust or low-pressure heater is operated as a condenser, and where the radiation takes the place of a cooling tower.

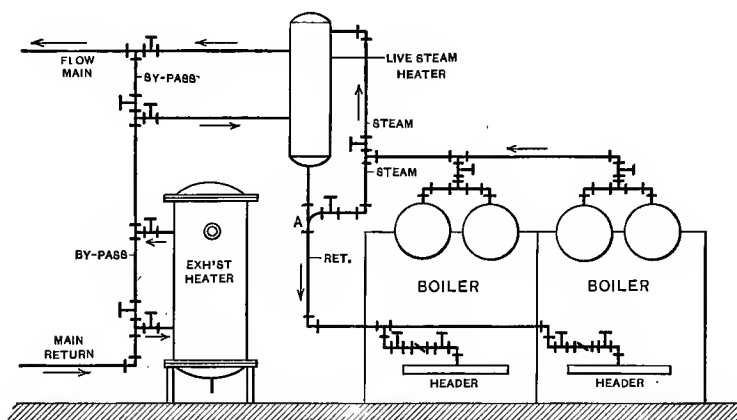


Fig. 77 Diagram of Connections for Live Steam and Exhaust Heaters.

In addition to this, the plant is usually provided with a regular condensing outfit for use during the non-heating season and for supplementing the action of the heater when needed.

The advantages of this arrangement are greater economy in the generation of power, and a wide temperature range under which the water may be circulated, due to the low pressures and temperatures existing in the heater.

Higher temperatures may be secured at any time, when needed, by diverting varying proportions of the water through a live-steam heater connected as shown in Fig. 77, or by passing the en-

tire volume of water through the secondary heater and varying the temperature by throttling the steam supply as already noted.

A typical lay-out for a system of this kind is shown in Fig. 78. It will be seen by examination that both heater and condenser are so connected with the exhaust main, hot-well and dry-air pumps that they may be operated independently if desired.

When the amount of exhaust steam exceeds the condensing capacity of the heater, under the desired vacuum, valve A may be closed and B opened, and the surplus steam from the heater allowed to pass into the condenser through valve C, which may

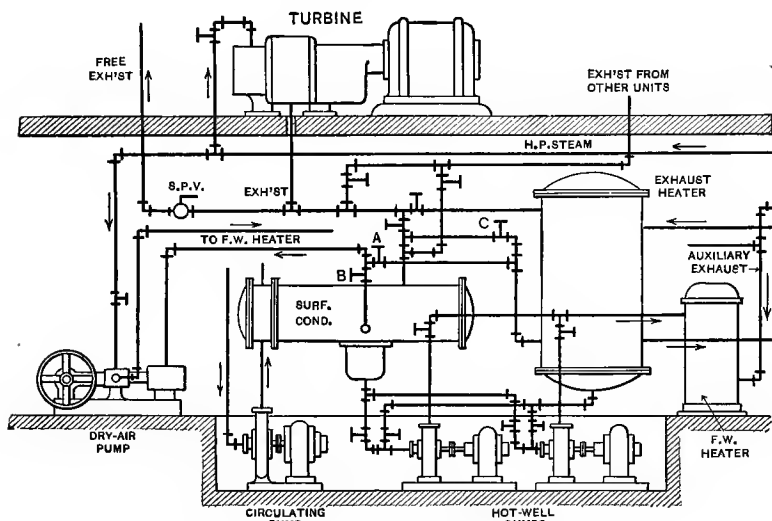


Fig. 78. System of "Vacuo" Hot-Water Heating.

be adjusted to produce the required vacuum in the heater. Under these conditions a full vacuum may be carried in the condenser, regardless of that in the heater, and may receive the exhaust from other units if desired through an independent connection as shown.

The condensing capacity of a given heating system, in pounds of steam, is easily determined by computing the heat given off by the radiation per hour, in B. T. U., and dividing the result by the latent heat of steam at exhaust pressure.

CHAPTER III.

WORKING DATA FOR HOT-BLAST HEATING AND VENTILATION.

The special requirements of different buildings, as regards ventilation, have been considered in some detail in Chapter I., together with the general methods employed for the supply and removal of air in each case.

The present chapter deals with the selection and arrangement of ventilating equipment and the computations involved in proportioning the different parts for various requirements.

General data relating to the composition of the air and the methods employed for determining and maintaining definite standards of purity have been fully covered in other books and will not be repeated here.

Fans.

120. Types Employed.—Two distinct types of fans are used in ventilating work, known as *centrifugal* and *propeller* fans, each having its own particular field although they may overlap in some cases.

121. Centrifugal Fans.—These include the regular paddle wheel blower or exhaustor, the multivane fan, and the cone fan.

The first of these is the standard type and is used more widely than any other although the multivane fan is rapidly coming into general use owing to its smaller size for a given capacity, and its higher efficiency.

Both of these types are positive in their action and easily give the pressures necessary to force the air through long ducts at high velocities.

They are used almost exclusively for air supply in all classes of large buildings and for exhaust ventilation also in the case of shops, factories, office and municipal buildings, or wherever the vent flues and ducts offer considerable frictional resistance.

Cone fans are often employed in schools and churches where large volumes of air are moved under comparatively low pres-

tures. They are usually made to discharge into a large chamber from which distributing ducts lead to different parts of the building.

122. Propeller Fans.—These are adapted to conditions like those shown in Fig. 24, where the air is drawn from large gathering chambers and discharged directly outboard against a low resistance as in the case of churches, schools, and hospitals. The smaller sizes are extensively used for local ventilation in rooms of various kinds. Propeller fans are light in construction as compared with the centrifugal type and are capable of handling large volumes of air with a small expenditure of power when operated under the conditions for which they are designed.

123. Fan Drives.—Ventilating fans may be driven either by steam engines, turbines, or electric motors, according to circumstances. When the exhaust can be utilized in the heater, an engine or turbine is usually more economical to operate where it is necessary to vary the fan speed to any great extent during different periods of the day. Where electricity is generated on the premises, motors are usually employed for all fan work, especially where ventilation is provided throughout the year regardless of heating, and where the exhaust from an engine would be wasted during the summer. Again, the matter of convenience and quietness of action often leads to the use of motors even at an increased cost of operation.

124. Motors for Ventilating Work.—As it is frequently desirable to connect the motor directly with the fan shaft a direct current is preferable in ventilating work. When this is not available, and an alternating current must be employed, it becomes necessary to use a belted motor in order to obtain the required speed reduction.

Shunt motors on constant potential circuits are best adapted to fan work, being self-regulating and running at a sufficiently constant speed for all practical purposes.

125. Speed Regulation.—This is obtained by the use of a regulating rheostat, which is similar in principle to a starting rheostat although somewhat different in construction. Starting rheostats, however, should never be used for speed regulation, as they are usually very compact and have a relatively small surface exposed

to the air. They do not heat much during the short time required for starting, but if allowed to carry the armature current constantly they are liable to burn out. Standard regulating rheostats for armature resistance give a variation in speed of about 50 per cent, and are commonly used for motors of less than 10 or 15 horse power. They have the disadvantage, however, of poor economy, because of the energy lost in heat.

126. Regulating by Varying Strength of Field.—Regulation by this means is generally accomplished by placing a rheostat in the field circuit. In this way the current passing through the field coils may be changed, and the field strength varied as desired.

The coils for armature and field resistance are usually combined in the same rheostat, so that moving the lever toward the right increases the speed of the motor by first cutting out the resistance in the armature circuit. A further movement of the lever throws an increasing resistance into the field circuit, thus weakening it and causing the speed to increase still more.

A speed variation of 100 per cent is commonly obtained with rheostats of this kind.

127. Regulation of Alternating-Current Motors.—An alternating-current motor of the induction type is a constant field machine, and therefore tends to run at a constant speed. A variable speed may be obtained either by changing the primary voltage of the motor or by introducing a resistance into the secondary circuit.

While induction motors are commonly used for running elevators, and similar purposes, where the reduction of speed is for short intervals, they are not so well adapted to ventilating work, where the requirements of speed regulation are much more severe.

128. Mechanical Regulation.—The continuous use of rheostats for speed regulation should be avoided when possible, it being better in the case of enclosed fans of the blower type to use throttling dampers in the airways, rather than depend entirely upon cutting down the speed. It is desirable at times, however, to reduce the capacity of a fan temporarily, as for an evening lecture or entertainment in the hall of a school building. In cases

of this kind the speed may be reduced for two or three hours by the use of a rheostat without a serious waste of energy.

If, in any given case, an alternating current must be used, a high-speed motor with belt connection is necessary.

129. Foundations.—It is of great importance to have the motor mounted upon a solid foundation, otherwise the vibration will cause sparking at the brushes and also prove objectionable on account of the noise produced, especially in buildings like schools, churches, etc.

Small motors may be supported in a variety of ways depending upon local conditions. They are often placed upon a shelf or bracket securely bolted to a brick wall, or in other cases hung from the roof construction of a building when used in connection with attic ventilating fans.

The larger sizes, such as are employed for driving blowers and exhausters, are generally placed upon solid brick or concrete foundations. If bricks are used they should be hard burned, and laid in a good quality of cement mortar; a substantial wooden frame should be placed over the brickwork to form an even support for the base of the motor to rest upon. If the machine is of the belt-driven type, it must be provided with a sliding bed-plate, having holding-down bolts and tightening screws for aligning and adjusting the belt.

The smaller sizes of direct-connected motors are usually bolted to the iron framework of the fan casing, while those of larger size are mounted upon a separate foundation and connected to the fan shaft by means of a flexible coupling.

130. Sound Insulation.—When it is necessary to reduce the vibration to a minimum, a sound deadener, made up of several layers of hair felt between two sheets of lead may be placed between the base of the motor and the foundation, before drawing down the anchor bolts.

In the case of small motors which are placed on brackets or other similar supports, thick rubber washers may be substituted for the felt. Still further insulation may be procured by covering the motor with a wooden box lined with felt and asbestos paper, care being taken to provide suitable openings for air circulation.

Another very satisfactory arrangement, which is adapted to engines also, is to build an 8-inch wall of brick or concrete

around the central foundation about 8 inches distant, and pack the space between with moist sand, as shown in Fig. 79.

131. *Capacity of Centrifugal Fans.*—(Paddle wheel type.)—The capacity of a fan of this type depends upon the diameter and width, the speed, and the resistance against which it operates.

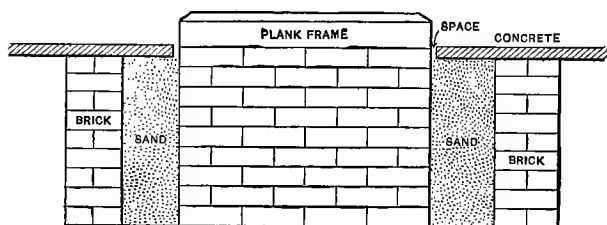


Fig. 79. Sound Deadening Foundation for Large Motors.

The following formulae are derived from an extended series of tests upon a standard blower of the paddle wheel type and give results corresponding very closely with those obtained in practice.

$$V = 2DNAB$$

$$N = \frac{V}{2DAB}$$

in which

V = cubic feet of air discharged per minute.

D = diameter of fan wheel, in feet.

N = revolutions per minute.

A = a factor, depending upon diameter of fan, (see Table XIX.).

B = a factor, depending upon resistance operated against, (see Table XX.).

The values of A , given in the table, are based upon the assumption that the width of the fan at the periphery is 0.5 the diameter. If the width is greater than this, increase A in the same proportion. For example, if the width is 0.6 the diameter for the make of fan used, multiply the tabulated value of A by

$$\frac{0.6}{0.5} = 1.2$$

TABLE XIX.

FAN CONSTANTS FOR DIAMETER.

Diameter of fan in feet.	Value of "A" in formulæ.
3	2.9
3½	3.9
4	5.0
4½	6.2
5	7.6
6	10.7
7	14.4
8	18.6
9	23.4
10	28.7

The resistance against which a fan operates varies with the friction in the ducts, which in turn, depends upon the length of run, the number of bends and the velocity of air flow.

In assuming the factor *B*, the following buildings are given as illustrating types in which the flue velocities and resistance to air flow are similar. These factors are based on experience, and will vary with the depth of heater, velocity of air flow, length of ducts and number of bends.

The lowest value (0.6) is for shops and factories employing deep heaters from twenty to twenty-eight pipes in depth with high air velocities of 1,800 to 2,000 feet per minute. The highest (0.9) applies to schools and churches with shallow heaters of open pattern and comparatively short ducts of large size. Between these extremes is a wide variety of conditions where the factor for resistance must be estimated from experience.

TABLE XX.

FAN CONSTANTS FOR RESISTANCE.

Type of building.	Value of "B" in formulæ.
Shops, foundries, factories, etc.	0.6 to 0.7
Office and municipal buildings, hotels. . . .	0.7 0.8
Schools, churches and halls.	0.8 0.9

The maximum speed at which a fan should run will depend within certain limits, upon the quietness of action it is desired to maintain. For ordinary conditions the following limiting speeds may be taken as corresponding well with practice, although with properly balanced fans, rigidly supported, they may be slightly exceeded without undue vibration.

TABLE XXI.

FAN SPEEDS FOR DIFFERENT PERIPHERAL VELOCITIES.

Diameter of fan, in feet.	Maximum revolutions per minute.		
	Shops and factories, (approx. peripheral vel. 5,500 ft. per min.).	Office and municipal buildings, (approx. peripheral vel. 4,500 ft. per min.).	Schools and churches, (approx. peripheral vel. 3,500 ft. per min.).
3	600	490	380
3½	500	420	320
4	450	360	280
4½	400	320	250
5	360	280	220
6	320	260	200
7	270	220	170
8	240	190	150
9	210	170	130
10	190	150	120

Table XXII. gives the cubic feet of air in round numbers, delivered by fans of different diameters for the speeds given in Table XXI. with values of B from Table XX.

TABLE XXII.

FAN CAPACITY FOR DIFFERENT CONDITIONS.

Diameter of fan in feet.	Cubic feet of air per minute for the speeds given in Table XXI.		
	Shops and factories, "B" = 0.6.	Office and municipal buildings, "B" = 0.7.	Schools and churches, "B" = 0.8.
3	6,200	5,900	5,300
3½	8,200	8,000	7,000
4	10,800	10,000	9,000
4½	13,400	12,500	11,200
5	16,400	14,900	13,400
6	24,600	23,300	20,500
7	32,700	31,000	27,400
8	42,800	39,600	35,700
9	53,000	50,000	43,800
10	65,400	60,000	55,600

Example.—What volume of air will be delivered by an 8-foot fan at 200 r. p. m. when used for the ventilation of an office building?

In this case

$$D = 8, N = 200, A = 18.6, B = 0.7$$

from which

$$V = 2 \times 8 \times 200 \times 18.6 \times 0.7 = 41,600 \text{ cubic feet per minute}$$

Example.—A shop building is to be supplied with 22,000 cubic feet of air per minute, what should be the diameter and speed of fan?

Looking in the column for "Shops and Factories" in Table XXII. it is found that this quantity comes between the limiting volumes given for 5-foot and 6-foot fans. As it is nearer the latter, the larger size will be chosen. The next step is to compute the required speed.

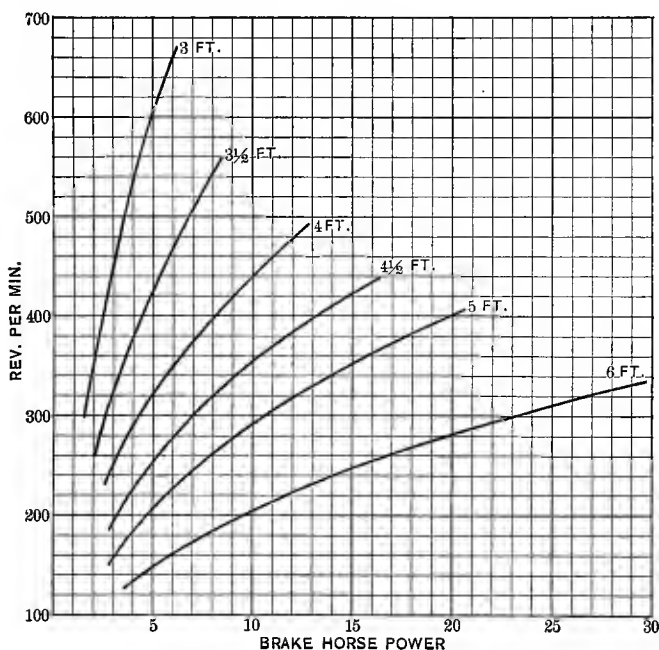


Fig. 80. Horse Power Chart for Centrifugal Fans.

Here we have

$$V = 22,000, D = 6, A = 10.7, B = 0.6$$

from which

$$N = \frac{22,000}{2 \times 6 \times 10.7 \times 0.6} = 286 \text{ r. p. m.}$$

In both of these examples the width of fan at the periphery has been assumed equal to 0.5 the diameter, which corresponds closely to the average fan of standard make.

132. Power for Driving Fans.—The power required for driving a fan varies with a number of governing conditions, hence for-

mulae applying with any degree of accuracy are somewhat cumbersome to use.

The curves given in Figs. 80 and 81 have been plotted from tables of horse powers and include the effects of speed, air volume, efficiency, etc., and are to be used in connection with the tables and formulae already given for capacity at different speeds.

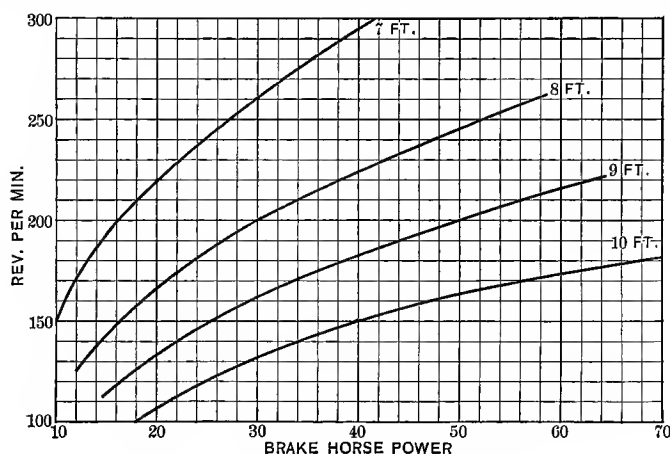


Fig. 81. Horse Power Chart for Centrifugal Fans.

Horse powers taken from the curves should be multiplied by the following factors to offset the effect of resistance:

*TABLE XXIII.

HORSE POWER FACTORS.

Value of "B."	H.P. factor.	Value of "B."	H.P. factor.
0.60	0.70	0.80	0.90
0.65	0.75	0.85	0.95
0.70	0.80	0.90	1.00
0.75	0.85		

* For use with curves in Figs. 80 and 81.

Example.—A factory requires a 7-foot fan running at a speed of 260 r. p. m. to deliver the required volume of air based on a resistance factor "B" of 0.65. What power of engine or motor should be provided? Fig. 81 shows that a 7-foot fan requires

30 D. H. P. at this speed, which, corrected for the effect of resistance actually calls for $30 \times 0.75 = 22.5$ D. H. P.

The effect of partially closing the discharge outlet from a fan, or its equivalent, increasing the resistance, is to reduce both the volume of air moved and the power required for maintaining a given speed. This holds good until the outlet is entirely closed, and the only power consumed is that used in churning the air within the fan-casing and overcoming the friction of the bearings.

133. Action of Multivane Fans.—As fans of this type vary considerably in form and general proportion it is not possible to give working data which will apply to all makes with any degree of accuracy as in case of the paddle-wheel type.

In a general way, the cubic space required by a multivane fan will be approximately one-half to two-thirds that required by the older type of the same capacity. Also, a multivane fan having the same outlet area as a paddle-wheel fan will deliver the same volume of air against a given resistance when running at about 65 per cent of the peripheral speed, and will require approximately 80 per cent as much power. While fans of this type run more quietly at higher speeds they are actually operated in practice at lower peripheral velocities than the paddle-wheel fan, varying from 2,600 to 3,000 feet per minute in the case of schoolhouses and churches, to 3,600 or 4,000 feet in factory work. Under ordinary conditions, static pressures of 1, $1\frac{1}{2}$, and 2 inches of water column may be assumed in schools, office buildings, and factories, respectively, when determining size, speed, and power of fan from manufacturers' tables. Also, the best results are obtained with corresponding outlet velocities of about 1,600, 1,800 and 2,000 feet per minute.

In specifying fans with which the engineer is not familiar it is usually best to state the volume of air to be moved per minute, the static pressure against which the fan is to operate, and the limiting peripheral velocity, rather than to give the size and speed. The static pressure in any given case may be approximately determined by means of Table XXX., Paragraph 148.

134. Propeller Fans.—Tests show that when operated under the usual conditions of discharge ventilation, that is, connected with a system of ducts in which the velocity does not exceed

about 700 feet per minute, a propeller fan of good type will move the air a distance parallel to the shaft equal to approximately 0.7 the diameter of the fan per revolution. Also the best results are obtained with a linear velocity through the fan of about 1,000 feet per minute, which corresponds with the following revolutions per minute for fans of different size, as given in Table XXIV.

TABLE XXIV.

ACTION OF PROPELLER FANS.

Diameter of fan, feet.	Approximate r.p.m. for linear vel. of 1,000 feet per min.
2	700
2½	575
3	475
3½	400
4	350
5	275
6	225
7	200
8	175

The above rule expressed as a formula is $V = 0.55D^3R$, in which

V = cubic feet of air moved per minute.

D = diameter of fan, in feet.

R = revolutions per minute.

The power required for driving propeller fans under these conditions will vary from 0.15 to 0.2 D. H. P. per 1,000 cubic feet of air moved per minute.

Example.—What volume of air will be discharged per minute by a 5-foot fan at 270 r. p. m., and what horse power will be required to drive it?

$$V = 0.55 \times 5^3 \times 270 = 18,562 \text{ cubic feet per minute.}$$

$$\text{H. P. for driving the fan} = 18.562 \times 0.2 = 3.7.$$

Heaters for Fan Work.

135. Forms Employed.—The radiating surface used in connection with blower work may be either in the form of wrought-iron pipe or cast-iron sections designed especially for this purpose.

In either case, standard forms are generally employed so that the work of the engineer relates rather to their arrangement than to details of design.

136. Effect of Depth and Air Velocity.—The capacity of a heater is affected by the depth, the amount of heating surface, the velocity of air flow between the pipes or sections, and the difference in temperature between the heating surface and the air. The final temperature increases with the depth, while the heat transmission per square foot of surface is reduced.

Changes in velocity have the reverse effect; that is, increasing the rate of air flow reduces the final temperature, but increases the heat transmission or efficiency. Varying the temperature difference between the heating surface and the air affects both the final temperature and the efficiency.

137. Computing Size of Main Heaters.—The first step in computing the size of a heater is to determine the volume of air to be supplied and its initial and final temperatures.

The standard formula for computing the heat necessary in work of this kind is

$$H = \frac{V \times T}{55}$$

in which

H = B. T. U. required.

V = cubic feet of air warmed.

T = rise in temperature, in degrees F.

While the factor 55 varies slightly for different temperatures, it may be used without correction for the range usually covered in all heating and ventilating work. The above formula may be transposed, as below, to meet other requirements.

$$V = \frac{55H}{T}$$

$$T = \frac{55H}{V}$$

Example.—What quantity of heat will be required to warm 30,000 cubic feet of air per minute from zero to 70 degrees?

$$H = \frac{30,000 \times 70}{55} = 38,182 \text{ B. T. U. per minute.}$$

Example.—What volume of air will be required to transmit 1,000,000 B. T. U. per hour to a building, by raising its tem-

perature 20 degrees above that of the normal inside temperature of the building?

$$V = \frac{55 \times 1,000,000}{20} = 2,750,000 \text{ cubic feet per hour.}$$

Example.—A room is to be supplied with 5,000 cubic feet of air per *minute* for ventilating purposes at a temperature of 70 degrees. It is desired to bring in 100,000 B. T. U. per *hour* additional for warming purposes. At what temperature must the air be admitted to the room?

$$T = \frac{55 \times 100,000}{5,000 \times 60} = 18 \text{ degrees}$$

which added to the normal temperature of the room, calls for an admission temperature of $70 + 18 = 88$ degrees.

Example.—A shop building is to be supplied with 10,000 cubic feet of air per minute. An inside temperature of 60 degrees is to be maintained in zero weather, with an estimated heat loss by transmission and leakage of 800,000 B. T. U. per hour. At what temperature should the air be admitted and what will be the total heat quantity required per hour for both heating and ventilation?

$$T = \frac{55 \times 800,000}{10,000 \times 60} = 73 \text{ degrees}$$

which added to 60 degrees, the room temperature, makes a final temperature and total rise of $60 + 73 = 133$ degrees. Therefore,

$$H = \frac{10,000 \times 60 \times 133}{55} = 1,450,909 \text{ B. T. U. per hour.}$$

Example.—Suppose in the preceding case no outside air was taken in by the fan in zero weather, and the building was warmed by re-circulated air leaving the heater at a temperature of 140 degrees and returning at 50 degrees. What quantity of heat would be required and what volume of air would it be necessary to pass through the fan and heater?

In this case the conditions are the same as in direct heating, and the required heat is simply that necessary to offset the losses due to transmission and leakage.

In cooling 1 degree, a cubic foot of air gives off $\frac{1}{55}$ B. T. U.,

and in cooling $140 - 50 = 90$ degrees, it will give off $90 \times \frac{1}{55} = 1.6$ B. T. U. Therefore, $800,000 \div 1.6 = 500,000$ cubic feet of air per hour must be passed through the fan and heater under the assumed conditions, which represent those commonly employed in practice.

138. Efficiency.—As previously stated, the efficiency of a heater depends upon its depth, the velocity of air flow, and the difference in temperature between the heating surface and the entering air. There are various formulae for determining the efficiency under different conditions, among which the following gives conservative results corresponding well with practice:

$$E = CD(T - t)$$

in which

E = efficiency of heater, in B. T. U. per square foot of surface per hour.

C = factor, depending upon depth of heater, (Table XXV.).

D = factor, depending upon air velocity through heater, (Table XXVI.).

T = temperature of steam or water in heater.

t = temperature of entering air.

Table XXV. includes heaters from 4 to 28 pipes in depth, which covers the usual range in heating and ventilating work.

TABLE XXV.

HEATER FACTORS FOR DEPTH.

Depth of heater, Factor "C." rows of pipe.	
4	2.50
8	2.24
12	2.03
16	1.86
20	1.72
24	1.60
28	1.50

Table XXVI. includes factors for air velocities of 800, 1,000, 1,200 and 1,500 feet per minute between the pipes. Velocities of 800 to 1,000 feet are commonly used in the case of schools and

other public buildings, while 1,200 or more are often employed in factory work where deeper heaters are required.

TABLE XXVI.

HEATER FACTORS FOR VELOCITY.

Velocity of air through heater, feet per min.	Factor "D."
800	3.6
1,000	4.1
1,200	4.5
1,500	5.0

139. *Final Temperature.*—This varies with the depth of heater, initial temperature and velocity of the air, and temperature of the heating surface. The following formula gives approximate results and may be used in connection with the formula for efficiency, previously given, when designing pipe heaters.

$$F = \frac{T - t}{G} + t$$

in which

F = final temperature of air passing through the heater.

T = temperature of steam or water in heater.

t = temperature of entering air.

G = factor, depending upon the depth of heater and velocity of air, (Table XXVII.).

(When t is below zero the formula becomes $F = \frac{T+t}{G} - t$.)

TABLE XXVII.

HEATER FACTORS FOR COMBINED DEPTH AND VELOCITY.

Depth of heater, rows of pipe.	Factor "G."			
	800 feet per min.	1,000 feet per min.	1,200 feet per min.	1,500 feet per min.
4	5.5	7.3	9.0	11.0
8	3.1	3.7	4.0	4.4
12	2.3	2.6	2.8	3.0
16	1.8	2.1	2.2	2.3
20	1.6	1.8	1.9	2.0
24	1.5	1.6	1.7	1.8
28	1.3	1.4	1.5	1.6

When it is desired to find the depth of heater for a given air velocity and final temperature, the formula may be written in the form $G = \frac{T-t}{F-t}$; then, having solved for G , its nearest value is found in Table XXVII. in the column corresponding to the assumed velocity, and the required depth of heater taken from column one.

The use of the preceding formulae and tables may be illustrated by computing the size of heater in some of the examples previously given. In the first of these it was found that 38,182 B. T. U. per minute were required to warm the air supply from zero to 70 degrees. Assuming a steam pressure of 2 pounds (220 degrees) and a velocity of 800 feet per minute

$$G = \frac{220-0}{70-0} = 3.1,$$

which, from Table XXVII., is found to call for a heater 8 pipes deep.

For the same conditions, with a heater of this depth (see Paragraph 138).

$$E = 2.24 \times 3.6 \times (220 - 0) = 1,760 \text{ B. T. U. per square foot of surface per hour.}$$

The total heat to be supplied is $38,182 \times 60 = 2,290,920$ B. T. U. per hour, from which the radiating surface is found to be $2,290,920 \div 1,760 = 1,302$ square feet.

In a following example a shop building was to be supplied with a certain volume of air at a temperature of 133 degrees in zero weather, requiring 1,450,909 B. T. U. per hour for raising it to that temperature.

Assuming a steam pressure of 10 pounds (240 degrees) and a velocity of 1,200 feet per minute, $G = \frac{240}{133} = 1.8$. Looking in Table XXVII. for a velocity of 1,200 feet, a heater between 20 and 24 pipes deep is called for. Using the latter, and solving for the efficiency, $E = 1.60 \times 4.5 \times 240 = 1,728$ B. T. U., from which $1,450,909 \div 1,728 = 840$ square feet of heating surface are required.

In the same example it was shown that with re-circulated air,

warmed from 50 to 140 degrees, it would be necessary to supply 800,000 B. T. U. per hour to offset the effects of transmission and

leakage. In this case, $G = \frac{240 - 50}{140 - 50} = 2.1$, which, from Table

XXVII., calls for 16 rows of pipe with an air velocity of 1,000 feet per minute. The efficiency under these conditions will be $E = 1.86 \times 4.1 \times (240 - 50) = 1,450$ B. T. U., requiring a heater containing $800,000 \div 1,450 = 552$ square feet of surface.

140. Cast-Iron Heaters.—The efficiency of cast-iron heaters, designed for hot-blast work, is very nearly the same as for pipe heaters and may be taken as ranging from 90 to 100 per cent, according to form.

In determining the depth, each 9-inch section will have practically the same effect upon the final temperature as four rows of pipe.

141. Area Through Heater.—In making up a heater care should be taken to so proportion the free area as to maintain the velocity of air flow assumed in the computations, even if it is necessary to increase the radiating surface beyond that called for. In the case of standard pipe heaters, about 0.4 of the overall or superficial area may be taken as free for the passage of air. The free area of cast-iron heaters is usually given in catalogues.

142. Supplementary Heaters.—When the warming is done by supplementary heaters placed at the bases of the flues, an efficiency of about 3.3 B. T. U. per square foot of surface per hour per degree difference between the heater and the average temperature of the air will be obtained. This applies to velocities of about 650 feet per minute, and, for average conditions with low pressure steam, gives a working efficiency of about $(220 - 85) \times 3.3 = 446$ B. T. U. per square foot of surface, or 450 B. T. U. in round numbers.

143. Pipe Connections.—Main heaters should be divided into six or eight separately valved sections in order to proportion the amount of surface in use to the requirements at different seasons of the year. Small heaters are sometimes divided into three sections with the surface in the ratio of 1, 2 and 3, which gives the effect of six sections of equal size by using the proper combinations.

When an engine is used for driving the fan certain sections should be valved for either live or exhaust steam, allowing from 30 to 40 square feet of condensing surface per horse power depending upon the type of engine employed. The exhaust portion of the heater should be divided into three sections, as considerably less surface is required in cold weather owing to its greater efficiency.

The returns from the different sections should be sealed from one another either by siphon loops or small traps to prevent the backing of condensation into the outer sections, due to a slightly reduced pressure in this part of the heater. Standard pipe heaters made up of vertical sections are particularly liable to this action unless the returns are sealed.

144. Pipe Sizes.—The steam supply to a main heater may be determined from Table XIII. by reducing the square feet of surface to its equivalent in direct, assuming the efficiency of the latter to be 250 B. T. U. For example, the ratio for a heater 12 pipes deep, with an air velocity of 800 feet per minute is approximately

$$\frac{1,500}{250} = 6.$$

For average conditions, with low pressure steam, and runs up to 100 feet in length, the sizes given in Table XXVIII. will usually be found ample.

TABLE XXVIII.

PIPE SIZES FOR MAIN HEATERS.

Square feet of surface in heater.	Steam pipe, inches.	Return pipe, inches.
150	2	1
250	2½	1¼
350	3	1½
550	3½	1½
700	4	2
1,300	5	2
2,100	6	2½
3,000	7	2½
4,500	8	3

145. Arrangement of Fan and Heater.—The best arrangement of fan and heater in any given case will depend upon the amount

and form of the available space and also upon the type of building and depth of heater. In general, the best results are obtained by drawing the air through the heater as in this case it is more evenly distributed over the entire radiating surface than when blown through at a high velocity. On the other hand, a greater volume of air must be handled by the fan for a given weight on account of the expansion due to heating. As the air quantity is usually based upon a given number of changes per hour or a certain amount per occupant, air volumes at normal inside temperatures are commonly used in all computations of this kind.

A typical arrangement of pipe heater and full-housed fan is shown in Fig. 82. The heater is supported upon iron beams

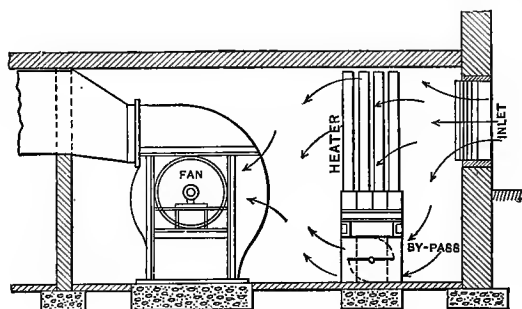


Fig. 82. Arrangement of Fan and Pipe Heater.

resting upon brick piers and is provided with a by-pass damper beneath it for accurate temperature regulation.

A common lay-out for a horizontal cast-iron heater is illustrated in Fig. 83. In the case of shops and factories, where floor space is limited, the arrangement shown in Fig. 84 may be used.

Air Distribution.

146. Methods Employed.—The general method of air distribution in different types of buildings has already been described.

In the present chapter working data relating to sizes of ducts and flues, the material employed, and details of construction will be considered.

147. *Size and Form.*—The mains are commonly based upon an assumed velocity, depending upon the type of building, while the branches, in most cases, are so proportioned as to have a combined carrying capacity equivalent to the main with which they connect. For example, four 8-inch pipes are equal in area to one 16-inch pipe, but when the frictional resistance is considered, it will take four pipes slightly over 9 inches in diameter to carry the same volume of air with the same drop in pressure per unit of length.

In the case of schools and similar buildings, where the air is delivered through the inlet registers at low velocities the duct and flue areas are proportioned upon a velocity basis through-

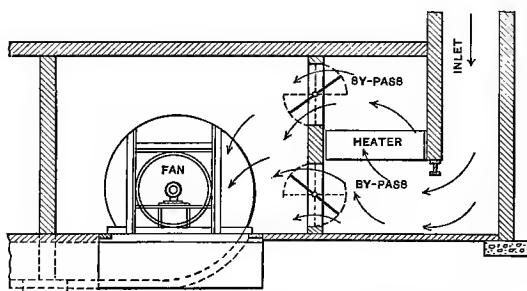


Fig. 83. Arrangement of Fan and Cast-Iron Heater.

out the entire system, reducing it gradually from the fan outlet to the registers. In buildings of this type it is customary to start with a velocity of 1,000 to 1,200 feet per minute in the main duct, reducing to 800 or 900 feet in the branches and to 600 or 700 feet in the vertical flues.

Air should not be admitted to banking rooms, dining rooms, class rooms, etc., at a velocity exceeding about 300 feet per minute over the gross area of the register, unless a diffuser is provided, in which case it may be increased to 400 feet per minute.

The vent flues in buildings of this kind may be made about 1.5 the area of the corresponding supply flues when the outward

air flow is by gravity, and 0.8 when an exhaust fan is used, both of which allow for a certain amount of outward leakage around doors and windows.

In office and municipal buildings, or wherever the flue area is restricted, higher velocities are employed, up to 1,400 or 1,500 feet in the mains, 1,200 feet in the branches and 1,000 feet in the flues. In the case of large supply uptakes, like that shown in Fig. 4, a velocity of 1,200 to 1,500 feet per minute may be allowed if the fan is located near the base of the flue.

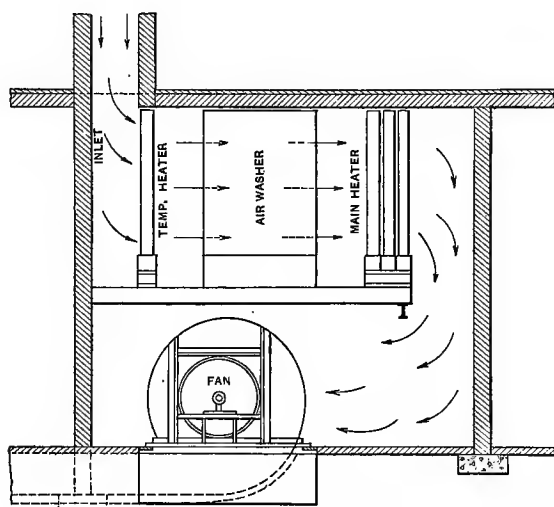


Fig. 84. Fan and Heater Arrangement where Floor Space is Limited.

In factories and shops velocities of 2,000 to 3,000 feet per minute are made use of in the trunk lines, reducing this in the branches to 60 or 70 per cent of the above. Another method frequently used is to make the area of the main duct the same as the discharge outlet of the fan and proportion the branches by means of the factors given in Table XXIX.

In this table, columns one and three give pipe dimensions, and columns two and four corresponding factors, the use of which is best explained by working a practical example.

TABLE XXIX.

FACTORS FOR EQUALIZATION OF RESISTANCE IN AIR DUCTS.

Diameter of round pipe or side of square pipe, inches.	Factor.	Diameter of round pipe or side of square pipe, inches.	Factor.
6	88	22	2,290
7	130	24	2,845
8	180	26	3,470
9	244	28	4,175
10	316	30	4,960
12	500	36	7,800
14	733	42	11,400
16	1,032	48	16,000
18	1,385	54	21,400
20	1,800	60	28,000

Example.—The discharge outlet of a fan has an area of 2,300 square inches, and the main duct is to be divided into two branches, what should be the diameter of each?

This area corresponds to a 54-inch round pipe, and the factor for this from Table XXIX. is 21,400. Then $21,400 \div 2 = 10,700$, which corresponds most nearly to the factor for a 42-inch pipe; hence, two 42-inch branches should be used.

Supposing in the above case it was desired to deliver the air through thirty outlets of equal size, what should be their diameter?

Proceeding as before, $21,400 \div 30 = 713$, which corresponds closely to the factor for a 14-inch pipe.

When the pipes are proportioned in this manner the trunk lines are not usually reduced each time a branch is taken off, but at intervals including several branches according to their size.

148. Computing Frictional Resistance.—As the size and speed of fan for a given capacity are often based upon the static pressure in the distributing ducts it becomes necessary to compute the frictional resistance in some cases in order to design this part of the equipment. While this may be estimated in certain classes of work, it is best with any unusual arrangement, especially in large buildings, to determine the resistance more accurately. Table XXX. gives the static pressure or resistance for round pipes, 100 feet in length, for different velocities of air-flow. For other lengths, the resistance will vary in direct proportion; that is, for a pipe 200 feet long, the resistance will be twice as great, and so on.

For velocities not given in the table, values may be interpolated with sufficient accuracy.

TABLE XXX.

STATIC PRESSURES.

Diameter of pipe, inches.	Static pressure in inches of water for pipes 100 feet long, at different velocities.			
	800 feet per min.	1,200 feet per min.	1,600 feet per min.	2,000 feet per min.
10	0.12	0.27	0.46	0.73
12	0.095	0.22	0.39	0.60
14	0.085	0.19	0.33	0.52
16	0.075	0.17	0.30	0.46
18	0.065	0.15	0.25	0.40
20	0.058	0.13	0.23	0.36
24	0.049	0.115	0.19	0.32
28	0.043	0.095	0.17	0.26
34	0.035	0.075	0.14	0.21
40	0.029	0.065	0.12	0.18
45	0.026	0.060	0.11	0.16
50	0.024	0.055	0.095	0.15
55	0.022	0.051	0.085	0.14
60	0.020	0.046	0.076	0.13
65	0.018	0.041	0.070	0.12
70	0.017	0.037	0.065	0.11
75	0.016	0.034	0.061	0.09
80	0.015	0.032	0.058	0.08

When the duct is rectangular in section it may be reduced to a circular form, having the same frictional area, by the formula

$$D = \frac{4 \times A \times B}{2(A+B)}$$

in which

D = diameter of an equivalent round duct and A and B the sides of the given rectangular duct.

The resistance offered by a 90-degree elbow will depend upon the radius of the inner curve or throat, and may be taken as follows:

TABLE XXXI.

RESISTANCE OF ELBOWS.

Radius, in diameters of pipe.	Length of straight pipe, in diameters, having the same resistance.
$\frac{1}{4}$	67
$\frac{1}{2}$	30
$\frac{3}{4}$	16
1	10
$1\frac{1}{2}$	6
2	4.3

The resistances produced by standard pipe heaters of different depths are given in Table XXXII., and for "Vento" cast-iron heaters in Table XXXIII.

TABLE XXXII.

RESISTANCE OF PIPE HEATERS.

Air velocity in feet per min- ute.	Resistance in inches of water column for heaters of different depths.					
	4 rows.	8 rows.	12 rows.	16 rows.	20 rows.	24 rows.
800	0.05	0.09	0.13	0.18	0.22	0.26
1,000	0.08	0.15	0.21	0.28	0.35	0.41
1,200	0.12	0.21	0.30	0.40	0.50	0.59

TABLE XXXIII.

RESISTANCE OF "VENTO" HEATERS.

Air velocity in feet per min- ute.	Resistance in inches of water column for heaters of different depths.					
	1 stack.	2 stacks.	3 stacks.	4 stacks.	5 stacks.	6 stacks.
800	0.37	0.70	0.103	0.135	0.167	0.200
1,000	0.59	0.109	0.160	0.221	0.262	0.313
1,200	0.84	0.157	0.230	0.303	0.376	0.449

In determining the pressure to be overcome in any given system the resistance is computed for the longest duct including bends, and to this is added the resistance due to the heater. If an air washer is used or the inlet duct is of considerable length, the effect of this should also be included.

When the main duct or trunk line is reduced at intervals it may be assumed that the smallest or extreme section of the duct is carried back to the fan independently, and the resistance computed on this basis.

Example.—The last section of an air duct is 18x24 inches in size, the extreme length of run 350 feet, and the air velocity through duct and heater 1,000 feet per minute. There are four elbows with radii equal to one-half the diameter, and a pipe heater is employed, 16 rows in depth.

What will be the total resistance against which the fan must operate?

Diameter of equivalent round pipe $= \frac{4 \times 18 \times 24}{2(18+24)} = 20.6$, which may be taken as 20 inches in round numbers. The resistance for 100 feet of straight pipe of this size with 1,000 feet velocity $= \frac{0.058+0.13}{2} = 0.094$ inches of water, (Table XXX.)

Effective length of run $= 350 + \frac{4 \times 30 \times 20}{12} = 550$ feet, (Table XXXI.)

Resistance due to pipe $= 0.094 \times 5.5 = 0.517$.

Resistance due to heater $= 0.280$, (Table XXXII.)

Total resistance $= 0.517 + 0.280 = 0.797$ inches of water column.

In some fan tables the pressure is expressed in ounces per square inch instead of inches of water. This change may be made by multiplying inches of water by 0.58, which, in the above case, gives a pressure of approximately $0.8 \times 0.58 = 0.464$ ounces per square inch.

149. Dampers and Deflectors.—Although the air distribution is proportioned to a considerable extent by the sizes of the ducts, it is necessary to provide deflectors at the principal junctions and dampers in the smaller branches for equalizing the flow to the different outlets. Register faces should be furnished either with valves for closing or shut-off dampers in the flues. In the case of mill work the air flow is regulated by means of especially designed deflecting dampers (see Fig. 58), when brick flues are employed.

With an overhead distribution, through metal ducts, each outlet should be equipped with an adjustable damper controlled by a pair of chains from the floor.

150. Material.—Underground airways are constructed of brick or concrete, smooth finished, with cement on the inside. When of any considerable length they should be lined with at least 2 inches of insulating material such as pressed cork or other block insulation solidly attached to the walls of the duct and finished with cement. The smaller branches in underground work are often made up of glazed sewer pipe. Overhead ducts, and the

vertical flues in buildings of wooden or semi-fire-proof construction are commonly made of galvanized iron, the thickness of metal depending upon the dimensions of the duct.

For ordinary work Table XXXIV. may be used, which gives the number according to the American gauge.

TABLE XXXIV.

GAUGE OF IRON FOR DUCTS.

Diameter of pipe or side of square duct, inches.	Gauge of iron, (American).
9	28
14	26
20	25
24	24
36	22
48	20
60	18
Above 60	16

For rectangular ducts of unequal dimensions the longer side may be 25 per cent greater than the diameter in the table for a given gauge of metal.

151. Registers and Grilles.—The material and design of the register or grille is usually determined by the architect, but the engineer should make sure that sufficient area is provided to limit the velocity and not impose too great a resistance upon the flow of air.

The figures previously given for the allowable velocity over the gross area of the register face are based upon a free area of 66 per cent of the gross area.

With registers of large size diffusing blades should be placed back of the grille work to equalize the flow of air over the entire surface, otherwise the velocity will be much greater at the top with the possibility of a reverse current through the lower part.

Air Washers.

152. Advantages.—The desirability of air “conditioning” has already been mentioned in connection with the ventilation of different types of buildings. The spray washer not only has the advantage of removing dust and other impurities from the air with a minimum amount of attention, but may also serve as a

means of humidity control and produce a certain amount of cooling effect in hot weather.

153. General Construction.—The standard type of washer consists essentially of a chamber provided with a sufficient number of spray heads for producing a suitable "water curtain," and an eliminator for removing any unevaporated moisture taken up by the air in passing through it. The general construction and arrangement of the spray heads or nozzles varies in different machines depending upon the form of spray desired. For the removal of dust only, a coarse spray resembling rain seems to give the best result, while for cooling purposes a fine mist is desirable on account of the greater evaporative effect.

154. Working Data.—The pressure required for producing the spray commonly runs from 10 to 30 pounds per square inch, depending upon the type of nozzle, and is furnished by a small centrifugal pump driven by a special motor. This arrangement is preferable to belting from the fan shaft, as it makes the pump independent under all conditions of operation, furthermore, the required power being small, only about 0.1 D. H. P. per 1,000 cubic feet of air per minute, the difference in efficiency is negligible.

For the best results, the air velocity through the washer should be low, and commonly runs from about 300 feet per minute in the smaller sizes up to 400 or 450 feet in the larger ones.

The space required for a standard equipment of this kind, per 1,000 cubic feet of air per minute capacity, is about 20 cubic feet, covering approximately 2 square feet of floor space. The general arrangement of an air washer, in relation to the fan and heaters, is shown in Fig. 84. The tempering coil or primary heater is placed between the inlet and washer and should be large enough to raise the temperature of the entering air above the freezing point in the most extreme weather.

155. Effect of Evaporation.—As the effect of passing air through a washer is to lower its temperature, due to evaporation, the secondary or re-heater should be of sufficient size to offset a drop of about 10 degrees, in addition to its normal capacity.

In other words, when a washer is used, add 10 degrees to the required rise in temperature when computing the size of heater.

Humidity.

156. Necessity of Control.—A working knowledge of the principles of humidity is necessary in the design of ventilating systems both for institutions and industrial plants.

In the former, lack of sufficient moisture produces irritation of the respiratory passages and makes it necessary to carry higher room temperatures in order to offset the cooling effect of increased evaporation from the body; while on the other hand, too much moisture is oppressive and produces a feeling of enervation. In the manufacture of textiles, and in other industrial processes, humidity control is an important factor and must be provided for. Data relating to evaporation and saturation are also necessary in determining air volumes and temperatures in the ventilation of paper mills, dye houses, etc., where large quantities of vapor are to be removed.

While the amount of moisture contained in a given space is independent of the presence of air, and depends entirely upon the temperature, it is more convenient in ventilating work to consider the moisture as contained in the air itself rather than in the space which it occupies.

157. Dew-Point, or Temperature of Saturation.—The maximum weight of moisture which a cubic foot of air will hold varies with the temperature, and the temperature of saturation for a given weight is called the *dew-point*, because any reduction in temperature below this will produce a certain amount of condensation which appears in the form of dew upon surrounding objects.

158. Absolute Humidity is the actual weight of moisture present and is usually expressed in grains per cubic foot.

159. Relative Humidity, at any given temperature, is the ratio of the weight of contained moisture to that which the air is capable of holding when fully saturated. For example, a cubic foot of saturated air at a temperature of 30 degrees contains 1.9 grains of moisture, while at 70 degrees it will hold 8 grains. Saturated air at the lower temperature has a relative humidity of

$\frac{1.9}{8} = 1$, that is, it is the same as the absolute humidity. Now

if the air be heated to 70 degrees without adding moisture, the

absolute humidity remains the same while the relative humidity falls to $\frac{1.9}{8} = 0.238$ or 23.8 per cent. This shows why the air in well ventilated buildings is so dry in winter time, even when the outside air has a high percentage of moisture, and illustrates the need of humidity control.

160. Measurement of Humidity.—Air is tested for humidity by means of wet and dry-bulb thermometers and the results read from a Hygrometric Chart, (see Fig. 85).

The dry-bulb thermometer indicates the normal temperature of the air, while the other, the bulb of which is kept constantly wet

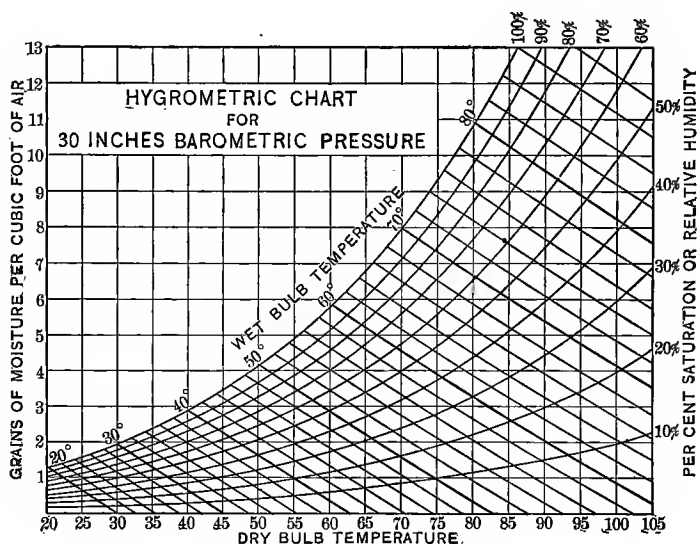


Fig. 85. Hygrometric Chart.

by means of a piece of wicking or other device, indicates a lower temperature due to the evaporation which takes place from its surface. It is evident that the lower the humidity the more rapid the evaporation from the wet bulb and the greater the difference in the readings of the two thermometers.

The chart given in Fig. 85 may be read with sufficient accuracy for ordinary work but for more careful results government tables should be used.

Example.—The wet and dry-bulb readings in a given room are 58 and 70 degrees, respectively, what are the relative and absolute humidities?

Follow up the vertical line for a dry-bulb temperature of 70 degrees until it intersects the oblique line for a wet-bulb temperature of 58 degrees. This in the present case falls on the curve for 50 per cent relative humidity or saturation. Following from the point of intersection to the left the absolute humidity is found to be 4 grains of moisture per cubic foot. When the point of intersection falls between the curves of saturation the distance must be estimated when making the readings.

The normal humidity of the outside air in the northeastern portion of the United States commonly ranges from 40 to 80 per cent. While the standard for occupied rooms varies from 60 to 70 per cent, this is likely to cause the moisture to gather upon the outside walls and windows in cold weather, hence, in practice, it is not best to exceed about 50 or 55 per cent on this account.

In the case of industrial plants the humidity must be governed by the character of the work.

161. Humidity Control.—Moisture may be imparted to the air in different ways; the more common being by means of an air washer, the use of submerged steam coils in evaporating pans located in the main airway, and the blowing of steam or a fine mist directly into the air currents as they pass to the building, or into the rooms themselves in special cases. When a washer is employed for filtering purposes it forms the simplest and most flexible method of controlling the humidity, as it is only necessary to maintain a certain relation between the temperature of the air leaving the washer and that of the spray water to secure any degree of humidity desired. In practice a constant air temperature is obtained by means of a duct thermostat attached to the tempering coil, while the spray water is controlled by a hot-water regulator which admits steam either to submerged coils or directly to the water through suitable mixing devices. This applies, however, only during the heating season when the entering air must be raised in temperature. In the summer, when the washer is operated for filtering purposes and the air is admitted at the outside temperature, too much moisture will often be absorbed unless means are taken to prevent it.

One method employed in this case is to simply flood the eliminator blades without using a spray, while another makes use of a solution of calcium chloride, 28-degree Baumé, in place of clear water which will thoroughly wash the air without either giving out or absorbing moisture. In certain branches of industrial work it is necessary to extract moisture from the air, or dehumidify it. Where maximum summer temperatures of about 90 degrees are permissible, sufficiently dry air may be obtained by using a solution of calcium chloride, 40-degree Baumé. A simpler method of reducing the humidity when city or well water at a temperature of 50 to 55 degrees is available, is to first cool the air, thus condensing a portion of the moisture, and then re-heat it, which reduces the relative humidity, as already described. For example, saturated air at 60 degrees when heated to 85 degrees has a relative humidity of about 45 per cent.

When for any reason air washers are not feasible, evaporating pans with steam coils, or direct sprays, may be employed, the amount of moisture being controlled automatically by means of humidostats placed in the room or in the air duct beyond the fan. While moisture may easily be introduced by means of steam jets, this method has the objection of increasing the temperature of the air, when heat is not required, and also of giving out an unpleasant odor unless special means are taken to prevent it.

Air conditioning forms a separate branch of engineering and important systems for this purpose should be designed by those having made a special study of the subject.

162. Removal of Vapor by Evaporation.—In the ventilation of paper mills and dye houses it is customary to pass large volumes of air through the rooms, which take up the moisture by evaporation and discharge it outboard. The carrying capacity of air used in this manner depends upon the initial and final temperatures and the outside humidity.

For example, a cubic foot of air at a temperature of 30 degrees, and 60 per cent relative humidity, contains 1.2 grains of moisture, (see Fig. 85). If this air be heated to 70 degrees, before delivering to a given room and then be discharged at the same temperature with a relative humidity of 80 per cent, each cubic foot will carry out $6.4 - 1.2 = 5.2$ grains of moisture. As there are 7,000 grains in a pound, $7,000 \div 5.2 = 1,346$ cubic feet of air must be

passed through the room for each pound of moisture removed under the above conditions.

In the case of paper machine rooms the air is discharged at a temperature approximating 100 degrees, with a relative humidity somewhat below saturation, say 90 per cent. With this data at hand the volume of air to be supplied for different outside conditions is easily computed. As the chart does not include the absolute humidity for higher temperatures and percentages of saturation, Table XXXV. is given for convenience. These results are taken from another chart but correspond closely with similar conditions in Fig. 85.

TABLE XXXV.

WEIGHT OF MOISTURE FOR DIFFERENT DEGREES OF SATURATION.

Normal or dry-bulb temperature, degrees.	Grains of moisture per cubic foot for different percentages of saturation.			
	70%.	80%.	90%.	100%.
85	9.0	10.2	11.4	12.7
90	10.3	11.7	13.3	14.6
95	11.9	13.5	15.5	17.0
100	13.5	15.6	17.7	21.0

Air-Cooling.

163. How Accomplished.—This is accomplished on a large scale in connection with the ventilation of hospitals, banking rooms, etc., in two ways: first, by means of a spray washer, and second, by mechanical refrigeration.

164. Cooling by Spray Washers.—This is the simplest and least expensive method, the cooling effect being due either to evaporation, or to direct contact with the water, depending upon the relative temperatures of the entering air and water and the amount of moisture in the former.

When the spray water is re-circulated, cooling is due entirely to evaporation, and at all other times when its temperature is at or above the dew-point temperature of the entering air. The difficulty experienced under these conditions is the increase in humidity necessary to secure an appreciable cooling effect. With re-circulated spray water the relative humidity of the air leaving the washer will average about 85 per cent, which, at room temperatures of 75 to 80 degrees, will prove oppressive. If sufficient

water is available from city mains or artesian wells to cool the spray to 3 degrees or more below the dew-point of the entering air, no evaporation will take place and cooling will be effected by direct transmission of heat from the air to the water. When this temperature difference exceeds 3 degrees, moisture contained in the entering air will be condensed, thus lowering the humidity in proportion to the difference in temperature and the volume of water passed through the washer.

The following tables, made up from charts published by the Warren Webster Company, show the approximate cooling effect which may be obtained under different conditions with their standard type of washer.

Table XXXVI. is for re-circulated water, and Table XXXVII. for deep well or city water at a temperature of 55 degrees, and both give the cooling effect for different temperatures and relative humidities of the entering air. In all cases of air cooling the matter of humidity should be taken into consideration when estimating the probable results.

Dew-point temperatures for use with these tables may be obtained from the chart in Fig. 85 for any given set of conditions as they correspond to the dry-bulb temperatures at 100 per cent saturation.

Example.—The wet and dry-bulb readings in a given case are 70 and 85 degrees respectively, what is the dew-point?

Follow down the oblique line toward the right for a wet-bulb temperature of 70 degrees until it intersects the vertical line for 85 degrees dry-bulb temperature; then horizontally toward the left to the curve for 100 per cent saturation, then vertically downward to the dry-bulb scale, where the reading is found to be 61 degrees.

TABLE XXXVI.

COOLING EFFECT WITH SPRAY WATER RE-CIRCULATED.

Relative humidity of entering air, per cent.	Temperature of entering air.		
	80 degrees.	90 degrees.	100 degrees.
30	11.2	12.4	13.6
40	9.0	10.0	11.0
50	7.5	8.2	8.6
60	6.0	6.4	6.6
70	4.5	4.7	4.8

TABLE XXXVII.

COOLING EFFECT WITH SPRAY WATER AT A TEMPERATURE OF 55 DEGREES.

Relative humidity of entering air, per cent.	Temperature of entering air.		
	80 degrees.	90 degrees.	100 degrees.
30	14.0	19.0	24.5
40	13.0	18.2	23.5
50	12.5	17.3	22.0
60	12.0	16.0	20.0
70	11.5	15.5	19.3

The above tables give the extremes, and between these, various results may be obtained, depending upon the quantity of cold spray water used and the minimum temperature maintained.

165. *Cooling by Refrigeration.*—When an ample supply of cold water is not available, so that the spray water must be re-circulated, it is impossible to reduce the air temperature an appreciable amount when the outside humidity is high without producing such a degree of saturation as to make the air very oppressive, and perhaps cause a deposit of moisture upon the walls and windows.

In cases of this kind it is customary to employ mechanical refrigeration, circulating the brine through coils over which the air is passed in the same manner as for heating. By this method the air is not only cooled, but the humidity may also be reduced as well.

Table XXXVIII. gives the tons of refrigerating capacity required per 1,000 cubic feet of air per minute for both cooling the air and condensing the vapor for different outside temperatures and humidities, with a constant inside temperature of 75 degrees and a relative humidity of 60 per cent.

For example, it will require a refrigerating plant of $(2.1+3.7) \times 10 = 58$ tons of ice melting capacity per 24 hours to cool and dehumidify 10,000 cubic feet of air per minute from a temperature of 85 degrees and 80 per cent humidity to the conditions stated above. In practice this should be increased about 2.5 per cent to cover the cooling of the vapor.

Sometimes a specially designed air-washer operated in connection with a refrigerating apparatus proves more economical than when the latter is used alone.

TABLE XXXVIII.

INSIDE TEMPERATURE 75 DEGREES, RELATIVE HUMIDITY 60 PER CENT.

Relative humidity of entering air, per cent.	Tons of refrigerating capacity per 1,000 cubic feet of air per minute for different temperatures of entering air.							
	80 degrees.		85 degrees.		90 degrees.		95 degrees.	
	For cooling air.	For condensing vapor.	For cooling air.	For condensing vapor.	For cooling air.	For condensing vapor.	For cooling air.	For condensing vapor.
60	1.7	1.0	2.1	1.7	2.5	2.8	2.9	4.2
70	1.7	1.7	2.1	2.8	2.5	4.2	2.9	5.6
80	1.7	2.7	2.1	3.7	2.5	5.2	2.9	7.0
90	1.7	3.4	2.1	4.7	2.5	6.3	2.9	8.4

166. *Computing Cooling Requirements.*—The general method of determining the volume of air and drop in temperature for maintaining a given difference between the inside and outside temperatures is practically the same as in heating, except the processes are reversed.

The sources of heat supply in this case are transmission from the outside through walls and windows, supplemented in varying degrees by the occupants, the lights, the operation of machinery, and industrial processes of various kinds.

Heat through transmission may be computed by the methods already given in Chapter II. In audience halls, and other rooms containing a considerable number of people, the animal heat given off must be taken into account, allowing about 400 B. T. U. per hour per occupant. Other sources of heat, such as lights, etc., should be estimated in each case according to conditions. Having determined the total amount of heat to be absorbed under maximum requirements, a sufficient volume of air must be provided to care for this by a rise in temperature within the range of the apparatus to be used.

Example.—The total heat to be absorbed per hour in maintaining a given temperature difference between the inside and outside of a building is 200,000 B. T. U. per hour. What air volume will be required with a temperature range of 10 degrees?

Using the formula $V = \frac{55H}{T}$ (see Paragraph 137)

we have $V = \frac{55 \times 200,000}{10} = 1,100,000$ cubic feet per hour or

$1,100,000 \div 60 = 18,300$ cubic feet per minute in round numbers. If mechanical refrigeration is to be employed the capacity of the machine may be obtained from Table XXXVIII. for the actual conditions of the problem, taking into account the power required for condensing the vapor as well as cooling the air.

The best relation to employ between temperature range and volume will depend upon the method of cooling and other local conditions. In any case the temperature of the entering air should not be so low as to cause uncomfortable drafts. In computing the temperature, a rise of 2 to 4 degrees between the fan and inlet registers should be allowed for.

CHAPTER IV.

CENTRAL PLANTS.

The following chapter includes only plants of small size, like those installed in connection with public institutions and small groups of buildings. Plants of this kind do not differ essentially from those in isolated buildings, except in the transmission of steam or hot water through underground mains, and what follows relates chiefly to this part of the equipment.

167. Advantages.—The principal advantages to be derived by the generation of power and heat for a group of buildings in a single plant are as follows: Saving in cost, both of equipment and of operation; convenience, due to the grouping of the apparatus at one point; and greater cleanliness, because the handling of coal and ashes is eliminated from the different buildings.

The economy of operating power and heating plants in combination is the same here as in a single building, and data given in the following chapters relating to the subject apply equally well to this class of work.

Systems of Heat Transmission.

168. Steam and Hot Water Compared.—Both steam and forced hot water are used in central heating, the advantages and disadvantages of each being much the same as already described in previous chapters. So far as the utilization of exhaust steam is concerned the results are practically identical whether it is condensed in the radiators or in a special heater located in the power house. While each system has certain features which may commend it more strongly in one location than another, it may be said in general, that either is adapted to the average group of buildings.

The relative elevation of the buildings to be heated with regard to the power house is sometimes a governing factor in the selection of a system. If certain buildings are too high the static pressure upon the apparatus will exceed the working limit and

the equipment must be divided into units, as in very tall buildings, or steam must be employed. In other cases, where one or more buildings are below the power house, difficulty in returning the condensation may turn the scale in favor of hot water.

While steam is capable of providing more general service in the way of cooking, heating water for laundry purposes, etc., it is often necessary, where the exhaust is employed for heating, to carry a separate high-pressure line for sterilizing, the operation of mangles, etc.; hence this advantage does not hold in the case of hospitals, or other institutions where service of this kind is required. The cost of laying the mains is somewhat less for steam than for hot water unless the elevations are such that extensive excavations are necessary for grading the pipes in order to secure the proper drainage. In hot-water heating the mains may follow the general contour of the surface at a suitable depth without reference to grades.

Radiation losses are greater in the case of steam, owing partly to the higher temperature carried, but more especially because the temperature is practically constant during the entire heating season.

Pipe Connections.

169. Steam Systems.—When exhaust or low-pressure steam is used the general method of running the mains and connecting them with the risers is the same as in isolated buildings. The power house, if possible, should be placed at a lower level than the other buildings in order that the condensation may flow back to the receiving tank by gravity. Return-line vacuum systems are particularly adapted to this class of work, especially when the units of the group are at practically the same level. If one or more of the buildings are necessarily located at a level below that of the power house the condensation must be returned to the main receiving tank by means of a special pump, either steam or electrically driven, according to circumstances. When high or medium pressure steam is available, a return trap works well under these conditions.

In case there is no surplus exhaust for heating purposes steam may be distributed to the different buildings at a pressure of 20 to 30 pounds, and reduced in each to the point required for heating. This arrangement furnishes steam for sterilizing and similar purposes, without running a separate line and also reduces the size of the supply main. In this case each building is provided with its own trap which discharges into a main return lead-

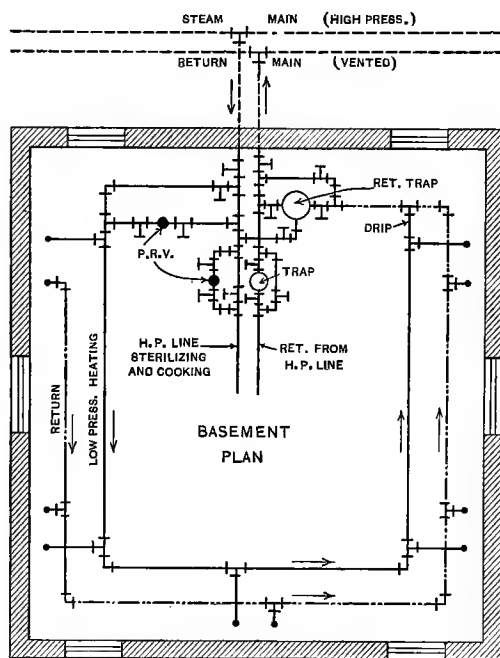


Fig. 86. Connections with Steam and Return Mains for Central Heating.

ing to a vented receiver in the boiler house. If the building is lower than the receiving tank, a return trap should be used, connected with the steam main outside the reducing valve, otherwise an ordinary steam trap will answer all purposes.

Piping arrangements for a system of this kind are shown in diagram in Fig. 86.

170. *Hot-Water Systems.*—The mains for hot-water distribution are usually run as shown in Fig. 87, while the connections

inside the buildings may follow any of the general methods already described. Although the circuit or loop system is adapted to single buildings, and in some cases to very small groups, the cooling effect is too great for general use in work of this kind. The rate of flow to the different buildings may be equalized by throttling the return branches, either by lock shield valves or by inserting disks with holes of suitable size.

Pipe Sizes.

171. *Steam Systems.*—The distributing mains used in central heating should be proportioned according to their length, the

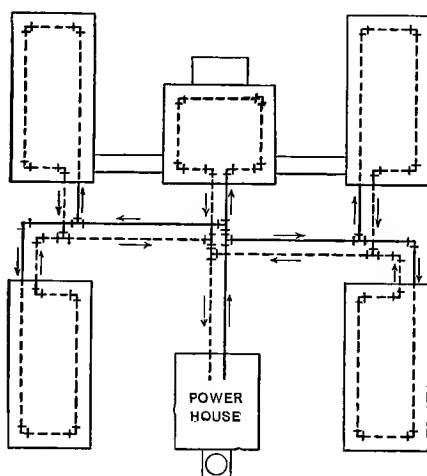


Fig. 87. Arrangement of Pipe Mains for Central Heating.

weight of steam to be carried, and the drop in pressure, rather than upon the amount of radiating surface to be supplied. There are several methods employed for this purpose among which the following, based upon *D'Arcy's* formula, has been used with satisfactory results. This is in such form that it may be employed in any case involving the flow of steam, whether for power, heating, or industrial purposes.

The formula written in various ways is given below

$$Q = c \sqrt{\frac{(p-p_1)d^5}{wl}}$$

$$W = c \sqrt{\frac{w(p-p_1)d^5}{l}}$$

$$d = \sqrt[5]{\frac{W^2 l}{c^2 w (p-p_1)}}$$

$$p-p_1 = \frac{Q^2 w l}{c^2 d^5}$$

in which

Q = cubic feet of steam per minute.

W = weight of steam per minute.

w = weight of a cubic foot of steam, at pressure p .

p = pressure of steam at inlet to pipe, called *initial* pressure.

p_1 = pressure of steam at outlet of pipe, called the *terminal* pressure.

d = diameter of pipe, in inches.

l = length of pipe, in feet.

c = a constant, depending upon the diameter of pipe.

TABLE XXXIX.

CONSTANTS FOR FORMULA FOR FLOW OF STEAM.

Diameter of pipe, inches.	Value of c .	Diameter of pipe, inches.	Value of c .
1	45.3	5	58.4
1¼	48.2	6	59.5
1½	50.3	7	60.1
2	52.7	8	60.7
2½	54.3	9	61.2
3	56.1	10	61.8
3½	57.1	12	62.1
4	57.8	15	62.6

Table XXXIX. gives values of c for different diameters of pipe, and Table XL. the fifth powers of the corresponding pipe sizes. As the computations are somewhat complex, tables of results are given with full directions for their use. These may be employed for all ordinary conditions and the formulae resorted to in special cases not covered by the tables.

Table XLI. gives the flow of steam in pounds per minute for pipes of different diameters, and with varying drops in pressure between the supply and discharge ends. These quantities are for pipes 100 feet in length; for other lengths the results must be corrected by the factors given in Table XLIII. As the length increases, the friction becomes greater and the quantity of steam discharged in a given time is diminished.

Table XLI. is computed on the assumption that the drop in pressure between the two ends of the pipe equals the initial pressure. If the drop is less than the initial pressure the actual discharge will be slightly greater than the quantities given in the

TABLE XL.

FIFTH POWERS OF PIPE SIZES FOR USE IN FORMULA.

Diameter of pipe, inches.	5th power.	Diameter of pipe, inches.	5th power.
1	1.00	5	31.25
1¼	3.05	6	77.76
1½	7.59	7	168.07
2	32.00	8	327.68
2½	97.60	9	590.49
3	243.00	10	1000.00
3½	522.90	12	2488.32
4	1024.00	15	10485.76

TABLE XLI.

FLOW OF STEAM IN POUNDS PER MINUTE FOR PIPES 100 FEET LONG OF DIFFERENT DIAMETERS AND WITH DIFFERENT DROPS IN PRESSURE.

Diameter of pipe, inches.	Drop in pressure, pounds.								
	¼	½	¾	1	1½	2	3	4	5
1	0.44	0.63	0.78	0.91	1.13	1.31	1.66	1.97	2.26
1¼	0.81	1.16	1.43	1.66	2.05	2.39	3.02	3.59	4.12
1½	1.06	1.89	2.34	2.71	3.36	3.92	4.94	5.88	6.75
2	2.93	4.17	5.16	5.99	7.43	8.65	10.9	13.0	14.9
2½	5.29	7.52	9.32	10.8	13.4	15.6	19.7	23.4	26.9
3	8.61	12.3	15.2	17.6	21.8	25.4	32.0	31.8	43.7
3½	12.9	18.3	22.6	26.3	32.5	37.9	47.8	56.9	65.3
4	18.1	25.7	31.8	36.9	45.8	53.3	67.2	80.1	91.9
5	32.2	45.7	56.6	65.7	81.3	94.7	120	142	163
6	51.7	73.3	90.9	106	131	152	192	229	262
7	76.7	109	135	157	194	226	285	339	390
8	108	154	190	222	274	319	402	478	549
9	147	209	258	299	371	432	545	649	745
10	192	273	339	393	487	567	715	852	977
12	305	434	537	623	771	899	1130	1350	1550
15	535	761	942	1090	1350	1580	1990	2370	2720

table, but this difference will be small for pressures up to 10 pounds, and can be neglected as it is on the side of safety. For

higher initial pressures, Table XLII. has been prepared. This is to be used in connection with Table XLI., as follows: First, find from Table XLI. the quantity of steam which will be discharged through the given diameter of pipe with the assumed drop in pressure; then look in Table XLII. for the factor corresponding to the assumed drop and the higher initial pressure to be used. The quantity given in Table XLI. multiplied by this factor will give the actual capacity of the pipe under the given conditions.

Example.—What weight of steam will be discharged through a 3-inch pipe, 100 feet long, with an initial pressure of 60 pounds and a drop of 2 pounds.

TABLE XLII.

VALUES CORRESPONDING TO HIGHER INITIAL PRESSURES TO BE USED IN CONNECTION WITH TABLE XLI.

Drop in pressure, pounds.	Initial pressure, pounds.						
	10	20	30	40	60	80	100
$\frac{1}{4}$	1.27	1.49	1.68	1.84	2.13	2.38	2.60
$\frac{1}{2}$	1.26	1.48	1.66	1.83	2.11	2.36	2.58
1	1.24	1.46	1.64	1.80	2.08	2.32	2.53
2	1.21	1.41	1.59	1.75	2.02	2.26	2.47
3	1.17	1.37	1.55	1.70	1.97	2.20	2.40
4	1.14	1.34	1.51	1.66	1.92	2.14	2.33
5	1.12	1.31	1.47	1.62	1.87	2.09	2.18

Looking in Table XLI. we find that a 3-inch pipe will discharge 25.4 pounds of steam per minute with a 2-pound drop. Then looking in Table XLII. we find the factor corresponding to 60 pounds initial pressure and a drop of 2 pounds to be 2.02. Then according to the rule given, $25.4 \times 2.02 = 51.3$ pounds, the capacity of a 3-inch pipe under the assumed conditions.

Sometimes the problem will be presented in the following way: What size of pipe will be required to deliver 80 pounds of steam a distance of 100 feet with an initial pressure of 40 pounds and a drop of 3 pounds?

We have seen that the higher the initial pressure with a given drop, the greater will be the quantity of steam discharged; therefore a smaller pipe will be required to deliver 80 pounds of steam at 40 pounds than at 3 pounds initial pressure. From Table

XLII. we find that a given pipe will discharge 1.7 times as much steam per minute with a pressure of 40 pounds and a drop of 3 pounds, as it will with a pressure of 3 pounds dropping to zero. From this it is evident that if we divide 80 by 1.7 and look in Table XLI. under "3-pound drop" for the result thus obtained, the size of pipe corresponding will be that required.

$$80 \div 1.7 = 47.$$

The nearest number in the table marked "3-pound drop" is 47.8 which corresponds to a 3½-inch pipe and is the size required.

TABLE XLIII.

FACTORS FOR PIPE LENGTHS OTHER THAN 100 FEET TO USE WITH
TABLE XLI.

Feet.	Factor.	Feet.	Factor.	Feet.	Factor.	Feet.	Factor.
10	3.16	120	0.91	275	0.60	600	0.40
20	2.24	130	0.87	300	0.57	650	0.39
30	1.82	140	0.84	325	0.55	700	0.37
40	1.58	150	0.81	350	0.53	750	0.36
50	1.41	160	0.79	375	0.51	800	0.35
60	1.29	170	0.76	400	0.50	850	0.34
70	1.20	180	0.74	425	0.48	900	0.33
80	1.12	190	0.72	450	0.47	950	0.32
90	1.05	200	0.70	475	0.46	1000	0.31
100	1.00	225	0.66	500	0.45		
110	0.95	250	0.63	550	0.42		

For lengths of pipe other than 100 feet, multiply the quantities given in Table XLI. by the factors found in Table XLIII.

Example.—What weight of steam will be discharged per minute through a 3½-inch pipe, 450 feet long, with a pressure of 5 pounds and a drop of ½ pound?

Table XLI., which may be used for all pressures below 10 pounds, gives for a 3½-inch pipe, 100 feet long, a capacity of 18.3 pounds for the above conditions. Looking in Table XLIII. we find the correction factor for 450 feet to be 0.47. Then $18.3 \times 0.47 = 8.6$ pounds, the quantity of steam which will be discharged if the pipe is 450 feet long.

Examples involving the use of Tables XLI., XLII. and XLIII. in combination are quite common in practice. The following shows the method of calculation:

Example.—What size of pipe will be required to deliver 90 pounds of steam per minute a distance of 800 feet with an initial pressure of 80 pounds and a drop of 5 pounds?

Table XLIII. gives the factor for 800 feet as 0.35 and Table XLII. that for 80 pounds pressure and 5 pounds drop as 2.09. Then

$$\frac{90}{0.35 \times 2.09} = 123$$

which is the equivalent quantity we must look for in Table XLI. We find that a 4-inch pipe will discharge 91.9 pounds and a 5-inch pipe 163 pounds. As 4½-inch is not commonly carried in stock we should probably use a 5-inch pipe in this case, unless it was decided to use a 4-inch and allow a slightly greater drop in pressure. In ordinary heating work with pressures varying from 2 to 5 pounds, a drop of ¼ pound in 100 feet has been found to give satisfactory results. With exhaust heating in the case of central plants, a less drop per unit of length is usually employed, depending upon the distance of the various buildings from the boiler house.

172. Hot-Water Systems.—The sizes of mains for forced hot-water heating may be determined as already described for isolated buildings, although somewhat lower velocities are employed when the distances are of considerable length. In general, the velocity should range from 420 to 480 feet per minute in the trunk lines and from 240 to 300 feet in the branches to the separate buildings, depending upon the size of pipe.

Anchors and Expansion Joints.

173. Methods of Taking Up Expansion.—In laying out a system of distributing mains the matter of expansion and contraction must be provided for, both in the case of steam and hot water.

This may be done in two ways; first, by the use of 90-degree offsets where the movement is taken up by swivel joints or by the spring of the pipe in the smaller sizes; and second, by means of expansion joints designed especially for this purpose. The frequent use of offsets not only adds to the cost of construction, but also increases the frictional resistance, especially in the case of

hot water, hence, it is usually necessary to supplement this method by the use of expansion joints, commonly of the *slip* or *diaphragm* types.

The arrangement of the buildings is usually such that most of the expansion may be cared for without the introduction of additional offsets as indicated in Fig. 87.

174. *Expansion Joints*.—Whenever long runs of straight pipe occur it is usually necessary to employ expansion joints. These

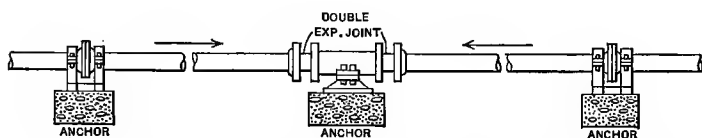


Fig. 88. Method of Pipe Anchorage when Using Double Expansion Joints.

should be of simple form, and so spaced that the maximum slipage will not exceed 5 or 6 inches. Under the ordinary conditions of low-pressure steam and hot-water heating the increase in length will not exceed 1.5 inch per 100 feet, so that expansion joints placed every 300 to 400 feet will answer the above requirements. With steam at 100 pounds pressure these distances should be reduced about 30 per cent. Data relating to the expansion of pipe under different conditions of temperature and

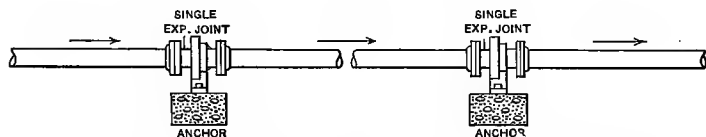


Fig. 89. Method of Pipe Anchorage when Using Single Expansion Joint.

pressure, and the proportioning of offsets and bends to secure the necessary flexibility, is taken up in detail in connection with Table LXXIX. in Paragraph 336.

175. *Anchors*.—In order to distribute the expansion evenly among the different offsets and joints it is necessary to anchor the pipe at regular intervals. There are various ways of doing this, one of which is shown in Fig. 88. In this case the pipe is

anchored midway between the joints, thus allowing expansion to take place in both directions from this point. Where two of these sections meet, a double expansion joint is provided, into which the ends of the pipes extending from the points of anchorage are inserted. The center, or stationary part of the joint is also anchored and service pipes or lateral branches taken from this point. Another method is to combine the anchor and expansion joint in a single fitting without the use of the separate or intermediate anchor. Such an arrangement is shown

in Fig. 89, where each unit consists of an anchor, a single expansion joint, and a service outlet if required. The expansion of each section in this case is in the same direction, as indicated by the arrows.

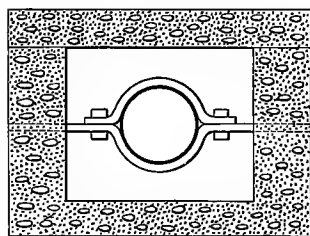
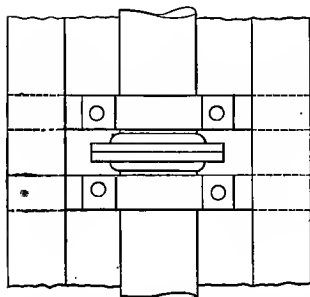


Fig. 90. Detail of Pipe Anchor.

An efficient and durable form of independent or midway anchor is illustrated in Fig. 90, and consists of a pair of heavy iron straps placed around the pipe at each side of a coupling or flanged joint, with the ends extending into the walls of the conduit at each side.

176. Special Expansion Joints and Anchors.—A double slip joint, combined with an anchor, is shown in Fig. 91. This is similar in construction to the piston rod and stuffing box of an engine or pump.

The barrel should be of brass to avoid rusting and adjustment is secured by means of bolts and nuts as indicated. An expansion joint of the diaphragm or variator type, also combined with an anchor and outlet, is shown in Fig. 92. In this case the expansion is taken up by flexible copper diaphragms of special form enclosed in an outer shell of cast iron. With this arrangement stuffing boxes are avoided and adjustments unnecessary, which is a matter of considerable importance in underground lines. Expansion

joints of corrugated copper pipe with supporting rings between the corrugations are also used in some cases.

177. Service Pipes should be taken off, when possible, at the expansion joints, for the reason that the main is practically stationary at these points, and also to avoid cutting it oftener than necessary.

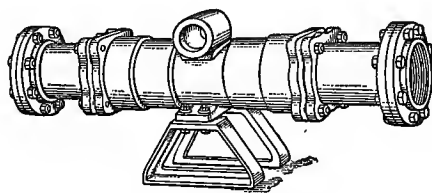


Fig. 91. Double Slip Joint.

If other connections must be made the points of take-off should be located near anchors.

Valves.

178. Type and Location.—All valves in work of this kind should be of the straightway gate pattern in order to reduce the resistance as much as possible. They should be placed in the trunk lines at the power house, in all branches where they leave the mains, and at such other points along the line as will enable sections to be cut out in case of repairs without laying up the entire system.

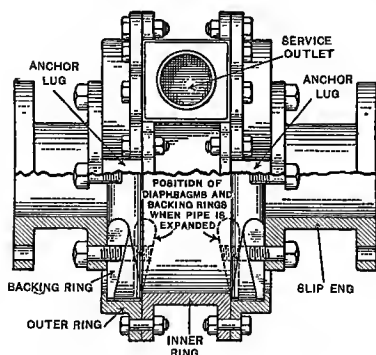


Fig. 92. Double Diaphragm Expansion Joint.

Conduits.

179. Materials.—In selecting the best type of conduit it is necessary to consider insulation, first cost and durability. The materials most commonly used in their construction are concrete, brick, vitrified tile and wood. Con-

crete is better adapted to this class of work than brick on account of its lower cost and also because it is more impervious to the leakage of water from the surrounding soil. Brickwork, how-

ever, is often used for the construction of manholes and chambers around valves and other fittings which must be accessible. Concrete and tile are the most durable materials and their use should be carefully considered when the plant is to be a permanent one.

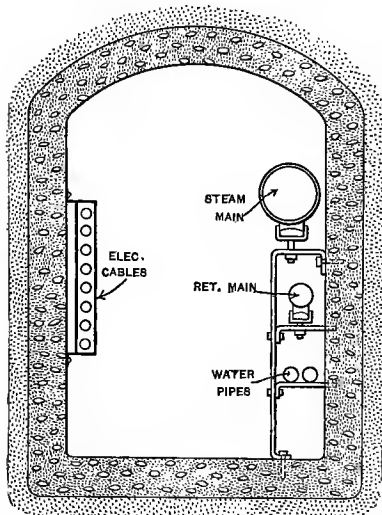


Fig. 93. Section through Concrete Pipe Tunnel.

The question is sometimes raised whether it is economy to construct a conduit, the life of which will probably exceed that of the pipe. This should, of course, be taken into account when the method of construction is such that the conduit cannot be opened without too great an expense when renewing the pipe.

180. *Tunnels* large enough for carrying the pipes and admitting workmen have many advantages in the way of inspection and repairs, but are very expensive to construct except for short distances. A section through a tunnel of this type is shown in Fig. 93.

181. *Types of Conduits.*—A simple form of closed trench or conduit is illustrated in Fig. 94. This is made of concrete, with an open top, and after the pipes and insulation are in place, is covered with slabs of the same material and the joints made tight with cement or tar.

A patented form of tile conduit, quite extensively used for long runs of pipe, is shown in Fig. 95. In this case the sections of tile are split lengthwise and the bottom half, forming the trench, laid first. The pipes are then put in place, being supported upon iron cradles resting upon concrete foundations and independent of the tiling. The cover or upper half is then put in place, the space around the pipe packed with asbestos sponge or other insulating material, and the joints made tight with

cement. Another form of patented conduit, especially adapted to wet locations, where it is liable to be submerged, is shown in Fig. 96. The lower half, which is of concrete, is first constructed and the pipes laid, after which the tile cover is put on and made tight by a cement dam at the sides and a special water-proofing poured while hot in the joints. The method of constructing the foundation and of preventing cracking and the inleakage of water are clearly indicated in the cut.

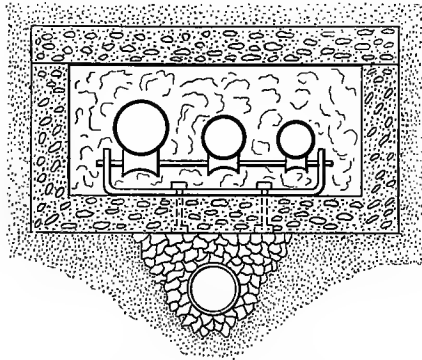


Fig. 94. Concrete Pipe Trench with Under Drain.

It is important that all pipe conduits be thoroughly underdrained with tile laid in coarse gravel or crushed stone, special care being taken to provide a free outlet for the drains at points sufficiently near together to keep the trenches free of water.

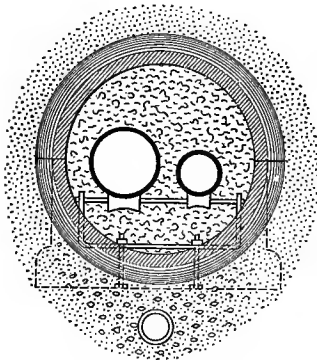


Fig. 95. Tile Pipe Conduit.

While concrete or tile conduits are the most durable, very good results have been obtained with those of wood when properly constructed and protected against moisture. A portion of a casing of this kind is shown in Fig. 97. The sections are made in lengths of 6 or 8 feet and are slipped over the pipe as it is laid. Solid turned logs are used for pipes up to 6 inches, and staves for the larger sizes. The shell varies in thickness from 2 to 4 inches according

to size, and the temperature carried in the pipes. It is strengthened with heavy galvanized wire and protected against moisture by a coating of asphaltum cement with a special creosote

preservative at the joints. The interior next the pipe is lined with bright tin. Another casing of this general type is illustrated in Fig. 98, and consists of a tin lining, outside of which are layers of asbestos, wood, corrugated paper, and finally an outer casing of wood staves coated with asphaltum. This form of construction has the advantage of being removable for repairs to the pipe.

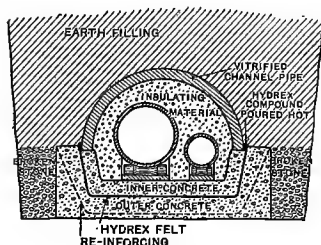


Fig. 96. Tile and Cement Pipe Conduit.

They may be constructed either of brick or concrete and should be provided with covers of sufficient strength to support the heaviest load to which they are likely to be subjected.

Depreciation of Pipe Lines.

183. Length of Life.—The life of a well-constructed conduit of cement should exceed that of the pipes which it carries, unless damaged by frost, the bursting of pipes, or other exceptional causes. For this reason the life of the conduit may

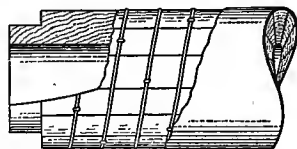


Fig. 97. Solid Wooden Pipe Conduit.

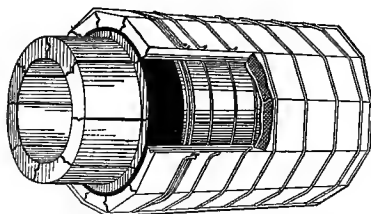


Fig. 98. Removable Wooden Pipe Conduit.

of piping packed in oiled shavings and enclosed in a conduit consisting of three layers of cypress boards 1 inch in thickness with air spaces between.

be considered as limited by that of the pipes unless it be of such form that the top may be removed without serious damage to the lower half. Table XLIV., giving the average life of pipes of different sizes used for hot-water heating is based upon a careful inspection of a system

The life of the pipe will, of course, be influenced by its quality, the presence of air upon the inner surfaces and moisture upon the outside.

TABLE XLIV.

LIFE OF PIPE MAINS.

Diameter of pipe, inches.	Length of life, years.
1½	21
2	22
3	32
4	35
5	38
6	41
8	47
10	53
12	55

The greater length of life given to the larger sizes of pipe is due to the heavier weight of the stock. The life of a wooden conduit depends largely upon the quality of the timber, the thoroughness of the under-draining and water-proofing, and the character of the soil in which it is laid. Conduits of this general construction have been known to last for 30 years, and several cases have been reported where they have been found in excellent condition after 10 or 15 years of service. On the other hand, they have deteriorated quite rapidly when of poor quality or laid in marshy soil.

One trouble experienced with wooden conduits comes from the great difference in temperature to which they are exposed during the heating and non-heating seasons, causing them in some cases to shrink and swell sufficiently to produce serious injury.

Table XLV. is for a cypress conduit of the construction already described in connection with Table XLIV., assuming the soil to be fairly dry and the trenches well drained.

TABLE XLV.

LIFE OF WOODEN CONDUITS.

Diameter of enclosed pipe, inches.	Life of conduit, years.
1½ to 2	16
3 " 8	25
10 " 12	30

Tile and cement conduits are practically indestructable so far as general deterioration is concerned, but may be affected by

exterior causes such as frost, failure of joints, bursting of pipes, etc.

184. Effect of Cost of Pipe.—In making up an estimate of the depreciation of any particular line, the relative cost of the pipe and conduit must be considered and the depreciation of each applied in its proper proportion.

Insulation.

185. Pipe Covering in Tunnels.—When considering the insulation of a pipe the entire covering, including the conduit, must be taken into account. In the case of brick or concrete tunnels the pipes should be protected with some form of sectional covering which has the property of resisting dampness as well as preventing the loss of heat. Although the tunnel may be dry in winter, when heat is on the mains, there is usually a period during the summer when moisture is likely to gather. The best grades of sectional covering have an efficiency of 75 to 85 per cent, which refers only to the covering on the pipes and not to the conduit as a whole. The effect of the tunnel will depend largely upon the tightness of the man-hole covers and the pipe openings into the various buildings. If these furnish sufficient area to produce a perceptible circulation of air through the tunnel the insulating effect due to the walls will not amount to very much, as the dead air space around the pipes is of the most importance. Therefore, in tunnel construction, the circulation of air through it should be kept at a minimum.

186. Pipe Covering in Conduits.—In well constructed conduits of tile or concrete, where the pipes are well packed with non-conducting material, the efficiency will be somewhat greater than for the sectional covering used in tunnel work, owing to its greater thickness.

Conduits of this type are usually made of such size that the thickness of insulation shall not in any case be less than 3 inches, which, in the best class of work, should bring the efficiency of the entire conduit up to 90 or 95 per cent.

With wooden conduits, the casing itself forms both the insulating material and the conduit, the whole construction forming

a unit. Recent tests upon a series of conduits of this type, the walls of which were 4 inches in thickness, showed an average efficiency of 95 per cent.

Wood casings are commonly made of such size as to give a dead air space around the pipe of 1 to 2 inches, the pipe being held in a central position by means of guides or runners.

CHAPTER V.

POWER AND STEAM REQUIREMENTS FOR DIFFERENT TYPES OF BUILDINGS.

187. *Power Requirements.*—The power requirements in the various buildings under consideration commonly include one or more of the following items: Electric lighting; elevator service; the driving of machinery; refrigeration; vacuum cleaning; electric current for miscellaneous motors, including those for cupola and forge blowers, ventilating fans, etc.; current for cooking and ironing; auxiliary equipment, including pumps and air compressors, and mechanical draft.

188. *Steam Requirements.*—Steam, in addition to that required for the generation of power, is also used for heating; ventilation; water heating; cooking; sterilizing; laundry work; vacuum cleaning, and various manufacturing purposes.

189. *Combined Power and Steam Requirements.*—These are arranged in Table XLVI., according to the buildings in which they are likely to occur and from which a schedule should be made as the first step in the design of an isolated plant. While only a part of these may be called for in buildings of small size a fairly complete list is convenient for purposes of reference, and selections can easily be made according to the case in hand.

TABLE XLVI.

POWER AND STEAM REQUIREMENTS.

OFFICE BUILDINGS.	
<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Ventilating fans	Hot water (lavatories)
Air cooling	
Auxiliaries	
Mechanical draft	
DEPARTMENT STORES.	
<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Refrigeration	Hot water (lavatories)
Cash system blowers	Cooking (lunch room)
Vacuum cleaning	
Ventilating fans	
Auxiliaries	
Mechanical draft	

MUNICIPAL BUILDINGS.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Ventilating fans	Hot water (lavatories)
Auxiliaries	
Mechanical draft	

HOTELS.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Refrigeration	Hot water (lavatories, baths, kitchen, laundry)
Vacuum cleaning	Mangles and dryers
Laundry and kitchen motors	Cooking
Electric cooking and ironing	
Ventilating fans	
Auxiliaries	
Mechanical draft	

HOSPITALS.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Refrigeration	Hot water (lavatories, baths, kitchen, laundry)
Vacuum cleaning	Mangles and dryers
Laundry and kitchen motors	Cooking
Electric cooking and ironing	Sterilizing
Ventilating fans	
Auxiliaries	
Mechanical draft	

THEATRES.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Ventilating fans	Hot water (lavatories)
Air cooling	
Vacuum cleaning	
Auxiliaries	
Mechanical draft	

TECHNICAL SCHOOLS.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Power for machinery	Ventilation
Ventilating fans	Hot water (lavatories and baths)
Auxiliaries	Cooking (lunch room)
Mechanical draft	

LOFT BUILDINGS.

<i>Power</i>	<i>Steam</i>
Lighting	Heating
Elevator service	Ventilation
Ventilating fans	Hot water (lavatories and industrial work)
Air cooling	Steam for various manufacturing processes
Power for machinery	
Auxiliaries	
Mechanical draft	

RAILROAD STATIONS.	
<i>Power</i>	<i>Steam</i>
Lighting	Heating (building and cars)
Elevator service	Ventilation
Refrigeration	Hot water (lavatories, car cleaning, etc.)
Air compression	Cooking (restaurant and lunch room)
Electric power for switches and signals	
Ventilating fans	
Auxiliaries	
Mechanical draft	

MACHINE SHOPS AND FOUNDRIES.	
<i>Power</i>	<i>Steam</i>
Lighting	Heating
Power for machine tools, etc.	Ventilation
Cranes and hoists	Hot water (lavatories)
Ventilating fans	Steam for special processes
Cupola blowers	
Auxiliaries	
Mechanical draft	

PAPER AND TEXTILE MILLS.	
<i>Power</i>	<i>Steam</i>
Lighting	Heating
Driving machinery	Ventilation
Elevators	Hot water (lavatories)
Ventilating fans	Steam for dyeing and other processes
Auxiliaries	
Mechanical draft	

In addition to the above there may be special requirements, both for power and steam, not included in the schedule, but which must be provided for in certain cases.

Computing Power Requirements.

190. Methods of Computation.—The following items have been taken from Table XLVI. and will be considered briefly under separate headings. Where power or steam requirements can be determined with any degree of accuracy for general conditions, simple methods are given for making fairly close approximations. While these are presented partly for purposes of illustration they are sufficiently accurate for much preliminary work. There are many requirements in the case of large buildings, however, which must be obtained from others in charge of special branches of the equipment. While the work of the steam engineer includes much detail of design, it is also made up to some extent of the assembling and connecting up of various specialties too numerous to consider in detail in the present volume. In deter-

mining the total power and steam requirements for any particular building accurate data relating to all equipment of this kind should be obtained from the manufacturers or from the architects or engineers having this part of the work in charge.

191. *Schedule of Power Requirements.*—

- (A) Driving Machinery.
- (B) Electric Lighting.
- (C) Elevator Service.
- (D) Refrigeration.
- (E) Vacuum Cleaning.
- (F) Cash System Blowers.
- (G) Cupola and Forge Blowers.
- (H) Laundry and Kitchen Motors.
- (I) Electric Cooking and Ironing.
- (J) Ventilating Fans.
- (K) Auxiliaries.
- (L) Mechanical Draft.
- (M) Air Compression.

In each case, unless otherwise noted, the power requirements will be reduced to *indicated horse power*, at the main engine, from which the weight of steam may be easily found.

The efficiencies assumed for engines, generators and motors in the following computations are somewhat lower than are ordinarily obtained in the case of large units, but are such as may be safely counted upon for the equipment usually found in isolated plants of small and medium size. When the actual efficiency of any machine to be employed is known it should be substituted for the assumed value in making the computations.

192. *Driving Machinery (A).*—In this case we will assume that the total horse power to be *delivered* to the main shaft of the mill or shop may be obtained either from the owners or the engineer in charge. Knowing the delivered or brake horse power (D. H. P.), the indicated horse power (I. H. P.), may be obtained by the formula:

$$\text{I. H. P.} = \frac{\text{D. H. P.}}{\text{M. E.}}$$

in which M. E. is the mechanical efficiency of the engine, and will

commonly run from 85 to 95 per cent at full load, depending upon the type and size.

Example.—A manufacturing plant requires 1,800 D. H. P.; what must be the I. H. P. of the engine if it has a mechanical efficiency of 90 per cent?

Solution:

$$\text{I. H. P.} = \frac{1,800}{0.9} = 2,000.$$

If the line shafting in a shop is to be motor-driven, then the total efficiency of engine, generator and motor must be taken

into account, and we have $\text{I. H. P.} = \frac{\text{D. H. P.}}{\text{T. E.}}$ in which T. E. =

the total efficiency, which may be taken as 65 to 75 per cent, under ordinary conditions, depending upon the size of the various units.

Example.—A shop, motor-driven, requires 700 D. H. P. at the main shaft; what should be the I. H. P. of the generator engine, assuming average efficiencies?

Solution:

$$\text{I. H. P.} = \frac{700}{0.7} = 1,000.$$

When the power required at the *machines* is given, the delivered power must be increased about 40 per cent to offset that consumed in turning the line-shaft and counter-shafting. When individual or small group drives are used this loss is practically eliminated, but on the other hand, the efficiency falls off so rapidly in motors of small size that the gain may be entirely offset under certain conditions. In practice the best results are usually obtained by a combination of individual and group driving.

Whatever the arrangement employed, a definite statement as to the power to be delivered at a given point should be obtained and the equipment proportioned accordingly.

193. Electric Lighting (B).—In order to determine the indicated horse power for lighting, a list of all the lamps must be made, together with the current in amperes required by each. The total current multiplied by the voltage will give the watts, from which the power required is easily computed, as will be shown later. Sometimes the candle-power and efficiency of a lamp are given instead of the rating in volts and amperes. In this case the

efficiency (in watts per candle-power) multiplied by the candle-power will give the total watts required for supplying it. In any case the first step is to reduce the electrical energy required for lighting to watts, and this divided by 746 will give the electrical horse power to be *delivered* by the generator. The indicated horse power of the engine is then found by dividing this quantity by the product of the generator and engine efficiencies. Placing the above in the form of an equation, we have

$$\text{I. H. P.} = \frac{I \times V}{746 \times GE \times ME}$$

in which

I = total current required, in amperes.

V = voltage.

GE = efficiency of generator = 90 to 95 per cent.

ME = efficiency of engine = 85 to 95 per cent.

Example.—What will be the required capacity of generator and indicated horse power of engine to supply the current for a 110-volt parallel system carrying one thousand 16-C. P. incandescent lamps at 0.6 amperes each, two hundred 32-C. P. lamps at 1.2 amperes each, 40 mercury vapor lamps at 3.0 amperes each and 100 arc lamps at 5.0 amperes each? Assume the generator efficiency to be 95 per cent, and the engine efficiency 90 per cent.

Solution:

$$1,000 \times 0.6 = 600$$

$$200 \times 1.2 = 240$$

$$40 \times 3.0 = 120$$

$$100 \times 5.0 = 500$$

$$\text{Total.} \dots\dots 1,460 \text{ amperes.}$$

Hence, the engine capacity is

$$\frac{1,460 \times 110}{746 \times 0.95 \times 0.9} = 242 \text{ I. H. P.}$$

and the generator capacity,

$$\frac{1,460 \times 110}{1,000} = 160 \text{ Kw.}$$

When making approximate estimates for preliminary plans where the number and type of lamps have not yet been worked out, the

following factors may be used, based upon the square feet of floor space to be lighted.

Incandescent lamps will require from 0.5 to 0.6 watts per square foot of floor space for general lighting in offices and halls, while drafting rooms and places requiring a more brilliant illumination take twice that.

These figures are for Tungsten lamps. In the case of carbon filament lamps of the older type, the lighting power per watt is about one-half the above.

For arc lighting the factors given in Table XLVII. may be used for closed lamps having opal globes.

TABLE XLVII.

Kind of room.	Watts per square foot of floor space.
Clothing store.....	1.30
Hall.....	1.00
Drafting room.....	2.00
Machine shop.....	0.75
Weave room.....	1.20

Example.—A building contains clothing stores on the first floor lighted by arc lamps, and offices above employing Tungsten lamps. Total floor space in stores, 5,000 square feet; in offices, 30,000 square feet. How many watts will be required for lighting?

Solution:

$$\begin{aligned} 5,000 \times 1.3 &= 6,500 \\ 30,000 \times 0.6 &= 18,000 \end{aligned}$$

$$\text{Total.....} 24,500 \text{ watts.}$$

Sometimes the capacity of the generator is given instead of the current requirements, and the indicated horse power of the engine computed from this by the equation:

$$\text{I. H. P.} = \frac{\text{Kw.} \times 1,000}{746 \times GE \times ME}$$

in which

Kw. = kilowatt rating of generator.

GE = efficiency of generator.

ME = efficiency of engine.

Assuming the lower efficiencies previously given and substituting

in the above, we find the indicated horse power of the engine required for each kilowatt capacity of the generator to be

$$\frac{1 \times 1,000}{746 \times 0.90 \times 0.85} = 1.75.$$

For units ranging from 100 to 500-kilowatt capacity, a ratio of 1.6 may be used, and 1.5 for larger sizes.

Example.—What should be the I. H. P. of engine to drive a 200-Kw. generator?

Solution :

$$200 \times 1.6 = 320.$$

194. *Elevators (C).*—The power required for running an elevator varies a good deal with the type, speed, and the general conditions under which it is operated. If specific data cannot be obtained in regard to the actual power necessary under the conditions of the problem, the following approximations may be used: In case of hydraulic passenger elevators assume a live load of 60 to 80 pounds per square foot of floor space, and weight of elevator as 100 to 125 pounds per square foot. Let the car be counter-balanced up to 75 per cent of its weight, and run at an average speed of 400 feet per minute, with an efficiency of 60 per cent. Elevators of this type require power only on the upward trip, as the car descends by its own weight.

The time required for making stops at the different floors, and such other time as the elevator may stand idle, is usually provided for by computing the power for continuous operation under full load and taking 50 per cent of the result.

The general formula for determining the horse power required for running an elevator is as follows:

$$\text{D. H. P.} = \frac{(W+w) \times S}{E \times 33,000}$$

in which

D. H. P. = delivered or brake horse power at machine.

W = weight of *live load*.

w = unbalanced weight of car.

S = speed in feet per minute.

E = efficiency.

In the case of hydraulic elevators the results given by this formula

must be divided by 2 and multiplied by 0.5 for the reasons stated above.

Working out several problems according to the assumptions noted, shows that from 0.4 to 0.5 horse power must be delivered to the elevator machine, for each square foot of floor space in the cars. The steam required for operating elevator pumps will be found in Table LXXIV., Paragraph 303.

In the case of electric elevators of the *drum* and *duplex* types, the efficiency commonly runs from 50 to 60 per cent. In practice, however, it is customary to use the lower figure which allows a wider margin of safety. The method of balancing these elevators differs from the hydraulic, in that the entire weight of the car and from 40 to 50 per cent of the maximum live load is counter-weighted. Examples involving these elevators, worked out for average conditions, show that from 0.3 to 0.4 delivered horse power are required per square foot of floor space. Assuming 0.35 as the average, and taking motor, generator and engine efficiencies as 0.86, 90 and 90 per cent, respectively, the required capacity of the generator engine is found to be

$$\frac{0.35}{0.86 \times 0.9 \times 0.9} = 0.5 \text{ I. H. P.}$$

per square foot of floor space in the cars.

The above gives *maximum* requirements for computing capacity of equipment. Operating costs should be based on *average* requirements, or about 0.7 of the maximum.

195. Refrigeration (D).—This is required wherever food is to be kept, as in hotels, hospitals, etc., and also in department stores for the cold storage of furs.

Large railroad stations and hotels are usually equipped for the manufacture of ice to be used in water coolers and for other purposes. Mechanical refrigeration is also used for air cooling under certain conditions as described in Paragraph 165.

As previously stated, the capacity of a refrigerating plant is usually expressed in "tons of refrigeration" or "ice-melting effect." For example, a 10-ton machine will produce the same cooling effect in 24 hours as the melting of 10 tons of ice; or, in other words, will extract the same amount of heat from the

circulating brine as would be required to melt 10 tons of ice into water at a temperature of 32 degrees. In case of the *compression* machine, working under average conditions, 1 indicated horse power at the steam cylinder will produce about 60 pounds of ice-melting effect per hour, or 0.75 ton per 24 hours. This will, of course, vary somewhat with the range of pressure and also with the size and type of machine, but in the absence of more exact data may be used for approximate results.

Another method in common use is to provide 1.5 indicated horse power per ton of refrigeration, which is slightly more than in the preceding case, the average of the two being about 1.4.

Example.—What I. H. P. of engine will be required to drive the compressor of a 20-ton ice machine?

Solution :

$$20 \times 1.4 = 28.$$

The delivered or brake horse power required at the machine may be estimated by assuming an engine efficiency of 0.85, which calls for $1.4 \times 0.85 = 1.2$ D. H. P. per ton of refrigeration.

If the machine is motor-driven the efficiency of the engine, generator and motor must be taken into account, which gives approximately

$$\frac{1.2}{0.7} = 1.8$$

I. H. P. per ton of refrigeration at the generator engine.

The above computations are adapted to the type of refrigeration commonly employed in hotels, hospitals, etc., where the cold storage rooms are cooled by circulation, or direct expansion coils supported along the walls, and also to the cooling of air for ventilating purposes. If power is required for the actual manufacture of ice, the results obtained by the above methods should be doubled.

196. *Vacuum Cleaning (E).*—The power required for vacuum cleaning depends upon the type and size of machine and the purpose for which, it is used, whether for carpets, bare floors, or walls. With machines of 4 to 8 sweepers capacity the required power at the exhauster commonly runs from 2.5 to 3 brake horse power per sweeper, the larger figure applying to the cleaning of bare floors, etc., where the volume of air handled is at a maximum.

In the case of motor-driven machines, the power required at the generator engine will run from 3.6 to 4.3 I. H. P. per sweeper, assuming the combined efficiency of engine, generator and motor to be 70 per cent. For purposes of approximation these figures may be taken as 4 and 5 I. H. P. respectively.

Cleaning systems of the steam jet type require from 200 to 250 pounds of steam per sweeper per hour.

197. *Cash System Blowers (F)*.—The power required for this purpose varies under different conditions and for accurate work should be obtained from the makers of the apparatus. In case the equipment is already installed the necessary power may be estimated by noting the rating of the blower motors and ascertaining if they are operated at their full capacity.

198. *Cupola and Forge Blowers (G)*.—The power for operating equipment of this kind is best obtained either from the makers or the engineers directly in charge of its installation. While data relating to this work may be taken from fan tables, the relations between air pressure and volume vary so widely from the conditions of ventilation that results obtained in this way are only approximate unless the exact requirements are at hand.

199. *Laundry and Kitchen Motors (H)*.—The power required for this purpose may be obtained by listing the various motors according to their ratings, or in case of a plant already in use, the current supplying them may be measured if it is considerable in amount and more accurate results are desired. Having determined the brake horse power to be delivered by these motors, the indicated horse power at the generator engine may be found by dividing this result by 0.7, the combined efficiency (assumed) of engine, generator and motors.

200. *Electric Cooking and Ironing (I)*.—When electricity is used for cooking, laundry work, etc., estimates must be made up from figures obtained from the makers of such apparatus or the current may be metered if the plant is already installed.

201. *Ventilating Fans (J)*.—The power for driving a ventilating fan may be determined from the size and speed of wheel and the resistance in the distributing ducts.

Data of this kind may be obtained from Table XXIII. in connection with the curves in Figs. 80 and 81, (Paragraph 132). This gives the power to be delivered at the fan. If an electric

motor is to be used the indicated horse power at the generator engine may be found by dividing the delivered or brake horse power by 0.7, the combined efficiency of engine, generator and motor.

202. Auxiliaries (K).—These include boiler feed and condenser pumps in the case of power plants, and circulating and vacuum pumps in connection with heating systems. Miscellaneous pumps, air compressors, etc., are also included under this heading.

When the feed and condenser pumps are driven by motors or steam turbines the required power may be based upon their ratings, or computed by the methods given in Paragraph 302. In the case of steam-driven pumps, of the piston or plunger type, it is customary to estimate the weight of steam required to operate them rather than the horse power. This for boiler feed, condenser, and vacuum pumps will vary in different plants, but for those of medium size may be taken as approximately 8 to 10 per cent of the steam supplied to the main engines.

The power for driving the circulating pumps in a hot-water heating system, operating under a 40-foot frictional head, will run from 1.5 brake horse power per 100 gallons per minute for the best type of turbine pump, up to 2 horse power for the cheaper grades of centrifugal pumps. For motor-driven outfits these figures should be divided by 0.7 to obtain the indicated horse power required at the main generator engine. If a direct-connected engine is used, divide by 0.8 the average efficiency of the type commonly employed for this purpose.

Power requirements for pumping water, etc., are treated quite fully in Paragraphs 302 to 304.

203. Mechanical Draft (L).—The power required for driving fans in connection with mechanical draft is small in comparison with the size of the plant. Assuming an average coal consumption of 4 pounds per hour per boiler horse power, it will require approximately $230 \times 4 = 920$ cubic feet of air for combustion, or $920 \div 60 = 16$ cubic feet per minute per boiler horse power to be supplied by the blast fan. The power required by the fan per cubic foot of air per minute to maintain a draft pressure of $1\frac{1}{4}$ inches of water column is approximately 0.0005 D. H. P. for forced draft and 0.0007 D. H. P. for induced draft. Assuming

an engine efficiency of 80 per cent for the type used, it will call for

$$\frac{0.0005 \times 16 \times 100}{0.8} = 1 \text{ I. H. P.}$$

for each 100 boiler horse power for forced draft, and

$$\frac{0.0007 \times 16 \times 100}{0.8} = 1.4 \text{ I. H. P.}$$

for induced draft.

Example.—A 1,200 horse power boiler plant is to be supplied with induced draft; what must be the I. H. P. of the engine for driving the fan?

Solution:

$$12 \times 1.4 = 17$$

204. *Air Compression (M).*—Compressors are usually rated in cubic feet of free air which they will compress per minute to a given higher pressure. When air is compressed, the work done during the process is converted into heat, which is taken up by the air. This causes a rise in temperature and increase in volume, thus adding to the work of compression. In case of the higher pressures it is customary to divide the process into stages, and pass the air through a cooler, containing pipes filled with water, between each stage. For pressures up to 60 pounds a single stage compressor is commonly used; from 60 to 200 pounds, two stages should be employed, and above that to 500 pounds, three stages. The theoretical horse power required to compress one cubic foot of free air per minute to different pressures is given in Table XLVIII.

TABLE XLVIII.
THEORETICAL HORSE POWER FOR COMPRESSING AIR.

Gauge pressure, to which air is compressed, pounds.	Horse power per cubic foot of air per min., single stage.	Horse power per cubic foot of air per min., two stage.
10	0.036	
20	0.063	
30	0.085	
40	0.104	
50	0.120	
60	0.134	
80		0.131
100		0.158
130		0.179
160		0.198
200		0.217

The efficiency of a single-stage compressor may be taken as about 60 per cent and that of a two-stage compressor as 75 per cent. Hence, to find the brake horse power required at the compressor, multiply the figures in the table by 1.66 and 1.33 for single-stage and two-stage machines respectively.

Reducing Power Requirements to Pounds of Steam.

205. Methods of Computation.—In order to compute the boiler power in a given plant it is necessary to reduce all power requirements to pounds of steam. This may be done by multiplying the total indicated horse power for all purposes involving the use of electricity, by the water-rate of the main generator engine. In the case of refrigerating machines, air compressors, power pumps, etc., driven by independent engines or turbines, the steam consumption should be computed separately, according to the water-rate of the special type of machine employed, and the result added to that of the main engine.

The water-rates of different types of engines are given in Tables LXIV. and LXV., Paragraph 263.

The water-rates of direct-acting steam pumps are based on the delivered or brake horse power and are given in Table LXXIV., Paragraph 303, which may be used in determining the steam requirements of hydraulic elevators or in other cases involving the use of pumps of this type.

Example.—A building requires 120 I. H. P. for lighting; 100 I. H. P. for electric passenger elevator service; 60 I. H. P. for driving miscellaneous motors; 40 I. H. P. for operating an independent steam-driven refrigerating machine, and 30 D. H. P. for a hydraulic freight elevator. The generators are equipped with medium-speed compound engines, the refrigerating machine with a simple belted high-speed engine, and the hydraulic elevator is operated by a compound non-condensing steam pump. What weight of steam will be required per hour by the power plant, assuming the auxiliary equipment to use an amount equal to 8 per cent of that required by the main generator engine?

From Table LXIV. the water-rates are found to be 21 and 30 pounds per I. H. P. per hour for medium-speed compound and simple high-speed engines, respectively, while Table

LXXIV. gives the steam consumption of a compound non-condensing pump as 65 pounds per D. H. P. per hour. From this data we have:

$$(120+100+60) \times 21 = 5,880$$

$$40 \times 30 = 1,200$$

$$30 \times 65 = 1,950$$

$$5,880 \times 0.08 = 470$$

Total:..... = 9,500 pounds of steam per hour.

206. *Procedure for Plants in Operation.*—The foregoing applies especially to the design of new plants, preliminary reports, etc. When it is necessary to determine the steam used for power purposes in plants already in operation it may be found by exhausting outboard and weighing or metering the water fed into the boilers; or if the plant is equipped with a surface condenser and the exhausts from all sources are connected with the same, the condensation may be measured instead of the feed water.

Computing Steam Requirements.

207. *Schedule of Steam Requirements.*—Steam requirements for other purposes than power are listed below in a similar manner.

(N) Heating.

(O) Ventilation.

(P) Hot Water.

(Q) Cooking.

(R) Sterilizing.

(S) Laundry Service.

(T) Steam Jet Vacuum Cleaners.

(U) Manufacturing Purposes.

208. *Heating (N).*—In case of a new building the total heat loss by transmission and leakage should be obtained by the methods given in Paragraphs 91 to 95, from which the maximum weight of steam may be found by the formula

$$W = \frac{H}{L}$$

in which

W = pounds of steam required per hour.

H = total heat loss from building, in B. T. U. per hour.

L = latent heat of steam at the pressure employed = 960
for 5 pounds gauge.

In case of a completed building, in which the radiation is already installed, the maximum weight of steam condensed per hour may be computed by the formula

$$W = \frac{(R \times E) + (r \times e)}{L} \times 1.2$$

in which

W = pounds of steam required per hour.

R = square feet of direct radiation.

r = square feet of indirect radiation.

E = 250 for low-pressure direct heating.

e = 400 for low-pressure indirect gravity heating.

L = latent heat of steam at the pressure employed = 960
for 2 pounds gauge.

With forced hot-water heating, multiply the radiating surface by 0.85 and proceed as for steam.

A more exact method is to trap the condensation from the entire system into a tank, determine its weight for a given period of time, and reduce to the conditions of zero outside temperature. When this is done the percentage of moisture in the steam entering the system should be measured by means of a calorimeter and its weight deducted from that of the returning condensation.

These methods are to be employed for determining the maximum steam requirements when computing the boiler horse power. In problems involving fuel consumption, the *average* condensation per square foot of surface for the heating season should be taken. This may be found approximately by first computing the maximum condensation, and multiplying the result by a factor depending upon the type of system and method of temperature regulation, as given below.

<i>Type of system</i>	<i>Factor</i>
Low-pressure steam, gravity system with hand valves...	0.80
Low-pressure steam, vacuum system with graduated valves	0.70
Forced hot water with hand regulation.....	0.60
Steam or hot water, with automatic regulation.....	0.55

209. *Ventilation (O).*—The weight of steam required for ventilation is based upon the volume of air supplied to the building per hour, and the outside temperature. This may be found by means of the formula,

$$W = \frac{V \times T}{55 \times L}$$

in which

W = pounds of steam required per hour.

V = cubic feet of air supplied per hour.

T = rise in temperature of air, in degrees.

L = latent heat of steam = 960 for 5 pounds gauge pressure.

For zero outside this becomes for 1 cubic foot of air

$$W = \frac{1 \times 70}{55 \times 960} = 0.0013$$

Example.—A building has an air supply of 2,000,000 cubic feet per hour. How many pounds of steam per hour at 5 pounds pressure will be required for ventilation in zero weather?

Solution:

$$2,000,000 \times 0.0013 = 2,600.$$

When it is desired to obtain the weight of steam required for ventilation during the entire heating season, an *average* outside temperature of 35 degrees may be assumed for New England and the North Atlantic States, which calls for $0.0013 \div 2 = 0.0007$ pounds of steam per cubic foot of air supplied.

210. *Hot Water for Lavatories and Baths (P).*—The steam necessary for heating water is easily determined if the quantity to be heated is known, together with the rise in temperature. This is given by the formula,

$$W = \frac{G \times 8.3 \times T}{960}$$

in which

W = pounds of steam at 5 pounds pressure.

G = gallons of water to be heated.

T = rise in temperature, in degrees.

If the water is to be heated from 50 to 130 degrees for bathing purposes, then for 1 gallon,

$$W = \frac{1 \times 8.3 \times 80}{960} = 0.7 \text{ pounds.}$$

The quantity of water to be heated in a given time can usually be obtained from the architect or owner of the building. If not, an approximation for hotels, apartment houses, hospitals, etc., may be made by assuming that from 20 to 30 gallons will be required for a tub bath, and that each tub will be used on an average of three times per hour during bathing hours.

Example.—A hotel has 100 public baths and 200 private baths. What weight of steam will be required per hour to heat the water from 50 to 130 degrees, assuming the public baths to be used three times each and the private baths once?

Solution: Total gallons of water used per hour = $(100 \times 25 \times 3) + (200 \times 25 \times 1) = 12,500$, from which the weight of steam required for heating it is found to be $12,500 \times 0.7 = 8,750$ pounds. The water required for lavatories in buildings of various kinds may be determined approximately by assuming that 1 gallon will be drawn each time the bowl is used, and estimating the frequency of use from the type of building. This multiplied by the number of bowls will give the gallons of water to be heated during the period taken. If steam is to be supplied to a system already in use the condensation from the heating coil may be measured as already described.

In the case of hotels and institutions the greater part of the hot-water supply is used during the morning and late evening, which include the hours for bathing. By providing a large storage tank the steam requirements for water heating may be distributed over an extended period, the length of which will depend upon the size of tank. When an instantaneous heater

is used the boiler capacity must be proportioned to heat the maximum quantity of water as needed.

211. *Cooking (Q)*.—Very little reliable data for general use is available upon this subject. If the steam supply is for a plant already in use, the present requirements may be determined by measuring the condensation. If the estimate is for a new plant, data as to the required steam supply should be obtained from the company furnishing the apparatus.

212. *Sterilizing (R)*.—Data regarding this must be obtained from the makers of the apparatus, or the condensation from a similar equipment may be measured.

213. *Laundry Service (S)*.—This includes steam for heating water, operating mangles, and for drying purposes. Here again an estimate of the steam required for the entire laundry outfit should be obtained from the makers of the apparatus to be installed. In the case of hospitals and similar buildings, from 5 to 6 pounds of steam per hour for each person in the entire institution has been found ample for heating the water for laundry and bathing purposes.

214. *Steam Jet Vacuum Cleaners (T)*.—A vacuum cleaning outfit of this kind requires from 200 to 250 pounds of steam per sweeper per hour, depending upon the vacuum carried and the class of work for which it is employed. High pressure steam should be used for this purpose.

215. *Manufacturing Purposes (U)*.—Conditions vary so widely that all data relating to the steam requirements for manufacturing purposes must be obtained from the owners or mill engineers, unless there is an opportunity to make tests and measurements at the plant, or at a similar one when the equipment to be supplied is new.

216. *Summary*.—As previously stated, whenever more exact data is at hand it should be substituted in the preceding formulae in place of the assumptions there given. In the absence of this, the average values found under the different headings may be used. For convenient reference the various methods employed for determining the power and steam requirements listed in Table XLVI. are given in abbreviated form in Tables XLIX. and L.

TABLE XLIX.

FORMULAE FOR DETERMINING THE INDICATED HORSE POWER OF MAIN ENGINE
FOR DIFFERENT POWER REQUIREMENTS.

(A) *Driving Machinery.*

$$(1) \text{ I. H. P.} = \frac{\text{D. H. P.}}{ME}$$

in which

I. H. P. = indicated horse power of engine when belted to main shaft.

D. H. P. = delivered or brake horse power required at main shaft.

ME = mechanical efficiency of engine, which may be taken as 0.85 to 0.95.

$$(2) \text{ I. H. P.} = \frac{\text{D. H. P.}}{TE}$$

in which

I. H. P. = indicated horse power of generator engine when the line shafting is motor driven.

D. H. P. = brake horse power required at main shaft.

TE = combined efficiency of engine, generator and motor, taken as 0.65 to 0.75.

(B) *Electric Lighting.*

$$(1) \text{ I. H. P.} = \frac{I \times V}{746 \times GE \times ME}$$

in which

I. H. P. = indicated horse power of generator engine.

I = total current required, in amperes.

V = voltage.

GE = efficiency of generator, taken as 0.90 to 0.95.

ME = efficiency of engine, taken as 0.85 to 0.95.

$$(2) \text{ Kw.} = \frac{I \times V}{1,000}$$

in which

Kw. = kilowatts.

I = amperes.

V = volts.

$$(3) \text{ I. H. P.} = \frac{\text{Kw.} \times 1,000}{746 \times GE \times ME}$$

in which

I. H. P. = indicated horse power of generator engine.

Kw. = rating of generator, in kilowatts.

GE = efficiency of generator.

ME = efficiency of engine.

$$(4) \text{ I. H. P.} = \text{Kw.} \times R$$

in which

I. H. P. = indicated horse power of generator engine.

Kw. = rating of generator, in kilowatts.

$R = \begin{cases} 1.75 & \text{for units of less than 100 kilowatt capacity.} \\ 1.60 & \text{for units of 100 to 500 kilowatt capacity.} \\ 1.50 & \text{for units above 500 kilowatt capacity.} \end{cases}$

(C) *Elevators.*

$$(1) \text{ D. H. P.} = \frac{(W+w) \times S}{E \times 33,000}$$

in which

D. H. P. = delivered or brake horse power required by elevator.

W = weight of live load.

w = unbalanced weight of car.

S = speed, in feet per minute.

E = efficiency.

$$(2) \text{ D. H. P.} = F \times 0.45$$

in which

D. H. P. = brake horse power required.

F = square feet of floor space in the cars of hydraulic elevators.

$$(3) \text{ D. H. P.} = F \times 0.35$$

in which

D. H. P. = brake horse power required.

F = square feet of floor space in the cars of electric elevators of the duplex and drum types.

$$(4) \text{ I. H. P.} = \frac{\text{D. H. P.}}{TE}$$

in which

I. H. P. = indicated horse power of generator engine for electric elevators.

D. H. P. = brake horse power required.

TE = combined efficiency of engine, generator and motor, taken as 0.7

Formulas (2) and (3) are for *capacity* of equipment.

For *average* operation, multiply the results by 0.7.

(D) *Refrigeration.*

$$(1) \text{ I. H. P.} = 1.4 \times TR$$

in which

I. H. P. = indicated horse power of engine for steam driven compressor.

TR = rating of ice machine, in tons of refrigeration per 24 hours.

$$(2) \text{ I. H. P.} = 2.8 \times TI$$

in which

I. H. P. = indicated horse power of engine for steam-driven compressor.

TI = rating of machine in tons of ice per 24 hours.

$$(3) \text{ D. H. P.} = 1.2 \times TR$$

in which

D. H. P. = delivered or brake horse power at machine.

TR = rating of ice machine in tons of refrigeration per 24 hours.

$$(4) \text{ I. H. P.} = 1.8 \times TR$$

in which

I. H. P. = indicated horse power at generator engine for a motor-driven machine.

TR = rating of ice machine in tons of refrigeration per 24 hours.

(E) Vacuum Cleaning.

$$(1) \text{ D. H. P. } = N \times 3$$

in which

D. H. P. = brake horse power for exhauster type of machine.
 N = number of sweepers.

$$(2) \text{ I. H. P. } = N \times 4.5$$

in which

I. H. P. = indicated horse power at generator engine, for motor-driven machine.

N = number of sweepers.

In case of steam jet machines, allow 250 pounds of steam per hour per sweeper.

(F) Cash System Blowers

$$(1) \text{ I. H. P. } = \frac{M}{TE}$$

in which

I. H. P. = indicated horse power at generator engine.

M = horse-power rating of motor.

TE = combined efficiencies of engine, generator, and motor, taken as 0.7.

(G) Cupola and Forge Blowers.

$$(1) \text{ I. H. P. } = \frac{M}{TE}$$

Symbols same as in (F)

(H) Laundry and Kitchen Motors.

$$(1) \text{ I. H. P. } = \frac{M}{TE}$$

Symbols same as in (F)

(I) Electric Cooking, Ironing, etc.

$$(1) \text{ I. H. P. } = \frac{I \times V}{746 \times GE \times ME}$$

in which

I. H. P. = indicated horse power at generator engine.

I = total current, in amperes.

V = voltage.

GE = 0.90 to 0.95.

ME = 0.85 to 0.95.

(J) Ventilating Fans.

See Figs. 80 and 81, and Table XXIII for D. H. P. required by fans.

$$(1) \text{ I. H. P. } = \frac{\text{D. H. P.}}{0.7}$$

in which

I. H. P. = indicated horse power at generator engine for motor-driven fans.

D. H. P. = brake horse power required by fan.

(K) Auxiliaries.

$$(1) W = E \times R$$

in which

W = weight of steam used by feed and condenser pumps.

E = weight of steam used by main engines.

R = factor, taken as 0.08 to 0.10.

$$(2) \text{ I. H. P. } = \frac{G}{40}$$

in which

I. H. P. = indicated horse power of engine for driving circulating pump in hot-water heating.

G = gallons of water circulated per minute.

$$(3) \text{ I. H. P. } = \frac{G}{35}$$

in which

I. H. P. = indicated horse power at generator engine for motor-driven pump.

(L) Mechanical Draft.

$$(1) \text{ I. H. P. } = \frac{1 \times \text{B. H. P.}}{100}$$

in which

I. H. P. = indicated horse power of fan engine for forced draft.

B. H. P. = boiler horse power.

$$(2) \text{ I. H. P. } = \frac{1.4 \times \text{B. H. P.}}{100}$$

in which

I. H. P. = indicated horse power of fan engine for induced draft.

B. H. P. = boiler horse power.

(M) Air Compression.

$$(1) \text{ D. H. P. } = \frac{C \times V}{E}$$

in which

D. H. P. = delivered horse power required at compressor.

C = theoretical horse power required to compress 1 cubic foot of air (see Table XLVIII).

V = cubic feet of free air compressed per minute.

E = efficiency of compressor = 0.60 for single-stage, and 0.75 for two-stage machine.

$$(2) \text{ I. H. P. } = \frac{\text{D. H. P.}}{0.7}$$

in which

I. H. P. = indicated horse power at generator engine for motor-driven compressor.

D. H. P. = delivered horse power required at compressor.

TABLE L.

FORMULAE FOR DETERMINING STEAM REQUIREMENTS FOR DIFFERENT PURPOSES.

(N) Heating.

$$(1) W = \frac{H}{960}$$

in which

 W = pounds of steam required per hour. H = total heat loss by transmission and leakage in B. T. U. per hour.

$$(2) W = \frac{(R \times 250) + (r \times 400)}{800}$$

in which

 W = pounds of steam required per hour. R = square feet of direct radiation. r = square feet of indirect radiation.

For weight of steam for entire season, see Paragraph 208.

(O) Ventilation.

$$(1) W = \frac{V \times T}{55 \times L}$$

in which

 W = pounds of steam required per hour. V = cubic feet of air supplied per hour. T = rise in temperature of air, in degrees. L = latent heat of steam at the pressure used.

$$(2) W = 0.0013 \times V$$

For a temperature rise of 70 degrees, with steam at 5 pounds pressure.

For weight of steam for entire season, see Paragraph 209.

(P) Hot-Water Heating.

$$(1) W = \frac{G \times 8.3 \times T}{L}$$

in which

 W = pounds of steam required. G = gallons of water to be heated. T = rise in temperature of water, in degrees. L = latent heat of steam at pressure used.

$$(2) W = 0.7 \times G$$

For a temperature rise of 80 degrees, with steam at 5 pounds pressure.

Items Q , R , S and U require special information and cannot be tabulated. Item T is included under E .

CHAPTER VI.

THE BOILER PLANT.

217. *Power Plant Equipment.*—The data given in connection with power plant equipment relates especially to matters of selection and arrangement, and to the methods of operation necessary for securing the greatest economy under existing conditions. Details relating to the theory and design of special machines and devices employed in this class of work have already been considered in a previous volume and do not come within the province of power plant design.

Selection of a Boiler.

218. *Factors to Be Considered.*—Among the governing factors in the selection of a boiler are pressure to be carried, size and number of units, available space, and cost. To these may be added details of construction, relating especially to accessibility for inspection, cleaning and repairs.

When the boilers are furnished by builders of established reputation, it is not usually necessary for the engineer to prepare detailed specifications. General requirements as to pressure, amount of heating and grate surface, type of furnace and setting are furnished the builder, from which he in turn prepares specifications for the approval of the engineer and which are submitted with his bid. This applies especially to water-tube and patented boilers, of which there is a great variety. In the case of fire-tube boilers of the horizontal type the engineer often furnishes specifications, particularly where any departure from standard construction is required.

219. *Types of Boilers.*—The boilers most frequently used in isolated plants are the horizontal return fire-tube and the standard makes of water-tube boilers. Vertical fire-tube, locomotive and marine boilers are also used to some extent in special cases. In the matter of fuel consumption for a given capacity there is very little choice in the different types of equal grade, and matters of adaptability and cost are the governing features in making a selection.

Return tubular boilers are rarely used for pressures over 150 pounds per square inch or for sizes much above 125 horse power, owing to the thickness of plate required, which in general, should not exceed $\frac{1}{2}$ or $\frac{5}{8}$ inch. Within these limits they are an efficient type and are especially adapted to low basements. The first cost of a return tubular boiler is somewhat less than for a water-tube boiler, which is sometimes a deciding factor.

For larger units and higher pressures some form of water-tube boiler is usually installed. Boilers of this type are also frequently employed in small and medium size plants on the score of greater safety, and special designs are constructed for locations where head room is limited.

Boiler Capacity.

220. Basis of Rating.—The capacity of a boiler is based upon the weight of steam which it will furnish in a given time under specified conditions. The standard of measurement adopted is the weight of dry steam evaporated per hour from and at 212 degrees, which corresponds to atmospheric pressure. The performance of a boiler operating under other conditions may be reduced to this standard by the use of Table LI.

TABLE LI.
FACTORS OF EVAPORATION.

Temperature of feed water, degrees F.	Steam pressure, by gauge.									
	80	90	100	110	120	130	140	150	160	200
32	1.222	1.225	1.227	1.229	1.231	1.232	1.234	1.236	1.237	1.240
40	1.214	1.216	1.219	1.220	1.222	1.224	1.226	1.227	1.229	1.232
50	1.204	1.206	1.208	1.210	1.212	1.214	1.215	1.217	1.218	1.221
60	1.193	1.196	1.198	1.200	1.202	1.203	1.205	1.207	1.208	1.211
70	1.183	1.185	1.187	1.189	1.191	1.193	1.194	1.196	1.197	1.200
80	1.173	1.175	1.177	1.179	1.181	1.183	1.184	1.186	1.187	1.190
90	1.162	1.165	1.167	1.169	1.170	1.172	1.174	1.176	1.177	1.180
100	1.152	1.154	1.156	1.158	1.160	1.162	1.164	1.165	1.167	1.170
110	1.142	1.144	1.146	1.148	1.150	1.152	1.153	1.155	1.156	1.159
120	1.131	1.133	1.136	1.138	1.140	1.141	1.143	1.145	1.146	1.149
130	1.121	1.123	1.125	1.127	1.129	1.130	1.132	1.134	1.136	1.138
140	1.110	1.113	1.115	1.117	1.119	1.120	1.122	1.124	1.125	1.128
150	1.100	1.102	1.104	1.106	1.108	1.110	1.111	1.113	1.115	1.118
160	1.090	1.092	1.094	1.096	1.098	1.100	1.101	1.103	1.104	1.107
170	1.079	1.081	1.083	1.085	1.087	1.089	1.091	1.092	1.094	1.097
180	1.069	1.071	1.073	1.075	1.077	1.079	1.080	1.082	1.083	1.086
190	1.058	1.060	1.063	1.065	1.066	1.068	1.070	1.071	1.073	1.076
200	1.048	1.050	1.052	1.054	1.056	1.058	1.059	1.061	1.063	1.066
210	1.037	1.040	1.042	1.044	1.046	1.047	1.049	1.051	1.052	1.057

Example.—A boiler supplied with feed water at a temperature of 70 degrees generates 4,000 pounds of dry and saturated steam per hour at a pressure of 180 pounds gauge. What is its equivalent evaporation from and at 212 degrees?

The factor for the above conditions from Table LI. is 1.2, hence the equivalent evaporation under standard conditions is $4,000 \times 1.2 = 4,800$ pounds of steam per hour.

The heat required to evaporate one pound of water from a temperature of 212 degrees into steam at atmospheric pressure is 970 B. T. U. (Table LXII.)

221. Horse Power Rating.—One boiler horse power represents the capacity to evaporate 34.5 pounds of water per hour from and at 212 degrees, which process requires $970 \times 34.5 = 33,465$ B. T. U. In practical work this is commonly taken as 33,000 which gives results on the side of safety.

222. Heating Surface.—What is commonly known as *heating surface* includes all plates and tubes exposed to hot gases on one side and water on the other. Surface coming above the water line and exposed to hot gases on one side and steam on the other is called *superheating surface*.

The effectiveness of the various heating surfaces depends upon their location and character, but in all computations relating to boiler capacity it is customary to assume a uniform value for the entire heating surface which shall represent a fair average under ordinary working conditions.

For power work an evaporation of 3 to 3.5 pounds of water per square foot of heating surface per hour may be taken, which

calls for $\frac{34.5}{3} = 11.5$ square feet per horse power in the first case

and $\frac{34.5}{3.5} = 9.9$ square feet in the second.

223. Builders' Rating.—Builders usually rate their boilers on a basis of 10 square feet of heating surface per horse power for water-tube boilers and 12 square feet for return tubular boilers, but as no uniform rule is followed in this respect the engineer should always specify the amount of heating surface required rather than the horse power. Some engineers call for a guar-

anteed evaporation under standard conditions as to feed temperature, steam pressure and coal consumption.

In the case of return tubular boilers it is well to state diameter and number of tubes for a given diameter of shell, as the efficiency is lowered by crowding them too closely together. Table LII. gives tube data as recommended by the Hartford Steam Boiler Inspection and Insurance Company.

TABLE LII.

DATA FOR RETURN TUBULAR BOILERS.

Diameter of shell, in inches.	Diameter of tubes, in inches.	Number of tubes.
48	3	44
54	3	54
54	3½	46
60	3	72
60	3½	64
66	3	90
66	3½	78
66	4	62
72	3	114
72	3½	98
72	4	72

224. *Efficiency.*—The efficiency of a boiler plant is made up of the combined efficiencies of the furnace and boiler and is expressed by the ratio,

$$\frac{\text{Heat absorbed by the water in the boiler per pound of coal, as fired.}}{\text{Calorific value of one pound of coal, as fired.}}$$

When oil fuel is used or mechanical appliances are provided for feeding the coal or creating a draft, the heat required per pound of fuel for this purpose must be deducted from that delivered by the boiler in the form of steam for useful purposes in order to obtain the net efficiency. The heat absorbed by the water in the boiler per pound of coal is equal to

$$\frac{970 \times W \times r \times f}{w}$$

in which

W = apparent weight of water evaporated, in pounds per hour.

x = quality of the steam.

f = factor of evaporation for the conditions of feed temperature and steam pressure during the test.

w = weight of coal burned, in pounds per hour.

The calorific value of coal varies with the kind and the locality from which it comes, and should be determined in each particular case for accurate results. For approximate work the following values may be used:

TABLE LIII.

CALORIFIC VALUE OF AMERICAN COALS.

Kind of coal.	Calorific value, in B. T. U. per pound.
Anthracite.....	13,200
Semi-anthracite.....	13,800
Semi-bituminous.....	14,700
Eastern bituminous.....	13,600
Western bituminous.....	12,300

In general, the efficiency will run from 50 to 70 per cent, averaging about 60 per cent in well designed and carefully operated isolated plants of good size. The above applies to boilers working under normal conditions.

When forced beyond the capacity for which they are designed the efficiency falls off somewhat, although not so much as was formerly supposed. Tests show that when a well designed boiler, proportioned for an evaporation of 3 pounds of water per square foot of heating surface per hour, is forced to evaporate twice that amount, the relative efficiency falls off less than 20 per cent.

Boiler Performance.

225. Results Included.—This relates to the various results obtained in the practical operation of steam boilers, such as rates of combustion and evaporation, coal per horse power, etc.

226. Rate of Combustion.—The weight of coal burned per square foot of grate per hour depends principally upon the kind of fuel, the type of furnace and strength of draft. The follow-

ing table gives about the average for different grades of coal burned under a natural draft. These figures, however, may be exceeded somewhat in large plants according to local conditions.

TABLE LIV.

RATE OF COMBUSTION WITH NATURAL DRAFT.

Kind of coal.	Pounds burned per square foot of grate per hour.
Anthracite buckwheat No. 1	9-12
Anthracite pea	12-15
Anthracite nut	14-18
Semi-anthracite, screenings	14-18
Semi-anthracite, run of mine	18-22
Semi-bituminous, screenings	18-24
Semi-bituminous, run of mine	18-24
Bituminous, slack	18-24
Bituminous, screenings	20-26
Bituminous, run of mine	20-28

With mechanical draft the rate of combustion may be greatly increased if desired, but is not usually carried much over 30 pounds in stationary plants of medium size. With certain types of stokers, however, the best results are obtained with small grate areas and high rates of combustion, but for average practice, with hand-fired furnaces, the figures given in Table LIV. should be generally followed.

227. Rate of Evaporation.—This depends partly upon the grade of fuel and partly upon the boiler and furnace efficiencies, which in turn are influenced by the character and arrangement of the heating surface and its relation to the grate area.

Table LV. gives the pounds of water evaporated per pound of coal for different calorific values and efficiencies within the usual range in isolated plants.

TABLE LV.

RATE OF EVAPORATION FOR DIFFERENT BOILER EFFICIENCIES.

Combined efficiency of boiler and furnace, per cent.	Pounds of steam from and at 212 degrees per pound of coal.			
	Calorific value of coal in B. T. U. per pound.			
	12,000	13,000	14,000	15,000
50	6.2	6.7	7.2	7.7
60	7.4	8.0	8.6	9.3
70	8.6	9.4	10.1	11.8

228. *Pounds of Coal per Horse Power per Hour.*—This is found by dividing 34.5 by the rate of evaporation obtained in any given case. Table LVI. has been prepared for the same conditions of boiler efficiency and calorific value of fuel as Table LV., and gives the pounds of coal required per boiler horse power per hour.

TABLE LVI.

COAL CONSUMPTION PER BOILER HORSE POWER PER HOUR.

Combined efficiencies of boiler and furnace, per cent.	Pounds of coal per boiler horse-power per hour.			
	Calorific value of coal in B. T. U. per pound.			
	12,000	13,000	14,000	15,000
50	5.6	5.2	4.8	4.5
60	4.7	4.3	4.0	3.7
70	4.0	3.7	3.4	3.2

Boiler Power for Different Purposes.

229. *Method of Obtaining.*—The boiler horse power for different purposes is easily obtained from the data given in Tables XLIX. and L., Paragraph 216.

230. *Boiler Capacity for Power.*—All power requirements should be reduced to indicated horse power. This multiplied by the water-rate of the engine, reduced to an equivalent evaporation from and at 212 degrees, and divided by 34.5, will give the boiler horse power required. Expressed as a formula, this becomes

$$\text{B. H. P.} = \frac{\text{I. H. P.} \times \text{W. R.} \times f}{34.5}$$

in which

B. H. P. = boiler horse power required.

I. H. P. = indicated horse power of engine to be supplied.

W. R. = water-rate of engine, (Tables LXIV. and LXV., Paragraph 263).

f = factor of evaporation for given conditions of feed temperature and pressure. (Table LI.).

Example.—What boiler capacity will be necessary to supply power for a factory requiring 500 I. H. P. at the engine? Power

to be furnished by a slow-speed simple condensing engine using 20 pounds of steam per I. H. P. per hour. Average feed-water temperature 60 degrees. Steam pressure 100 pounds gauge.

In this case we have

$$\text{I H. P.} = 500.$$

$$\text{W. R.} = 20.$$

$$f = 1.198.$$

which, substituted in the formula, calls for

$$\frac{500 \times 20 \times 1.198}{34.5} = 347 \text{ boiler horse power.}$$

This gives simply the boiler power for supplying the engine. If steam is required for other purposes such as the driving of pumps, heating, ventilating, etc., the capacity should be increased accordingly.

231. Boiler Capacity for Heating, Ventilation, Etc.—The general method in this case is the same as above described. All heating requirements are first reduced to pounds of steam per hour by means of data given in Table L. This multiplied by the proper factor of evaporation, and the result divided by 34.5 will give the boiler horse power. When exhaust steam is utilized for this purpose the amount used should be deducted from the total required in computing the boiler horse power, as described later.

Draft.

232. Method of Producing.—The necessary draft for burning the fuel in a boiler furnace is produced either by a chimney or by mechanical means. The first method, known as natural draft, is commonly employed in the average isolated plant, although it is sometimes strengthened by mechanical devices.

Mechanical draft is usually produced by a fan and when used alone is generally confined to manufacturing plants or to institutions having a separate power house. Most large buildings have sufficient height to produce the required draft without extending the chimney any great distance above the roof, and in any case the top of the stack must be at a sufficient elevation to carry off the gases of combustion. When the grade of fuel or type of stoker used calls for mechanical draft it is commonly employed in connection with a chimney in plants of this kind.

233. Strength of Draft Required.—The strength of draft required depends upon the grade of fuel, rate of combustion, and also upon the form and length of the various smoke passages. The resistance to be overcome is commonly considered as being made up of three parts; that between the ash-pit and furnace; between the furnace and smoke outlet, and between the smoke outlet and chimney.

Table LVII. gives common values for the first of these for different grades of fuel with average rates of combustion taken from Table LIV., and may be used in the absence of more definite information.

TABLE LVII.

DRAFT REQUIRED FOR DIFFERENT KINDS OF COAL.

Kind of coal.	Average rate of combustion, pounds per square foot of grate per hour.	Required draft between furnace and ash-pit, in inches of water.
Anthracite, buckwheat No. 3	10	0.40
Anthracite, buckwheat No. 1	12	0.30
Anthracite, pea	15	0.30
Semi-bituminous, run of mine	24	0.28
Bituminous, slack	24	0.22
Bituminous, run of mine	28	0.16

The draft pressure required for overcoming the resistance of the furnace and smoke passages will vary in different cases, but for average conditions, from 0.5 to 0.7 inches of water column may be assumed. In designing a chimney the draft necessary for overcoming the total resistance should be taken into account when making the computations.

234. Chimney Design.—This is usually included in the building contract, but the dimensions are commonly furnished by the engineer in charge of the steam plant. The requirements of a chimney are that it shall provide sufficient draft to overcome the resistances noted above and also carry off the gases of combustion. There are various formulae in use for the design of chimneys for power-plant work, but space does not allow of a discussion of these in detail. A simple method, which gives results corresponding well with current practice, is to compute the height by the formula,

$$H = \frac{d}{0.0056}$$

in which

H = height of chimney, in feet.

d = total draft required, in inches of water.

The flue area may be determined by use of Table LVIII.

TABLE LVIII.

HEIGHTS AND FLUE AREAS OF CHIMNEYS.

Height of chimney in feet.	Ratio of flue area to grate area.	
	Anthracite.	Bituminous.
100	0.143	0.166
125	0.125	0.143
150	0.111	0.125
200	0.100	0.111

Example.—A boiler plant has a total grate area of 80 square feet, and it is desired to burn 15 pounds of anthracite pea coal per square foot per hour. What should be the chimney dimensions?

Total draft = $0.30 + 0.6 = 0.9$ inches of water; calling for a

height of $\frac{0.9}{0.0056} = 161$ feet.

Area of flue = $80 \times 0.111 = 8.88$ square feet, which corresponds to a diameter of approximately $3\frac{1}{2}$ feet.

In designing a chimney, provision should be made for any forcing of the boilers which is likely to occur. This also applies to the equipment for mechanical draft.

235. Mechanical Draft.—Steel plate centrifugal fans are commonly used for this purpose, both the paddle-wheel and multi-vane types being adapted to this class of work. Two methods are employed, known as *forced draft* and *induced draft*. In the first of these the air is forced into the ash-pit under pressure, and in the second a suction is applied at the smoke opening in the boiler setting. Forced draft is somewhat cheaper to install owing to the smaller size of fan and connecting ducts, but does not produce such a uniform pressure over the entire grate as an induced draft. Furthermore, care must always be taken to shut off the pressure before opening the fire doors else flame and gases will be blown into the boiler room. Induced draft produces practically the same effect as a natural or chimney draft, but as the heated

gases pass through the fan, a larger volume must be handled, thus requiring an equipment of greater capacity.

236. Air Required for Combustion.—About 230 cubic feet of air at a temperature of 70 degrees are required for the combustion of each pound of coal burned under mechanical draft. Knowing the grate area and rate of combustion, the volume of air to be supplied in any given case is easily determined.

237. Fan Equipment for Forced Draft.—The method of computing the fan and power requirements for mechanical draft have been considered in detail in preceding volumes, hence only tabulated data will be given in the present case. Table LIX. relates to fans of the standard paddle-wheel type having a width equal to one-half the diameter. Table LX. applies to the "Sirocco" fan, which is of the multivane type. Both tables are based upon an air temperature of 70 degrees and a velocity of 2,000 feet per minute through the fan outlet. The horse power in each case represents the actual brake horse power required at the fan shaft. When selecting an engine, turbine or motor for driving the fan, an allowance of about 10 per cent should be provided for emergencies.

The upper line in each table gives the diameter of the fan wheel, and the second line the cubic feet of air moved per minute under the conditions of speed and pressure stated below.

238. Fan Equipment for Induced Draft.—When induced draft is employed certain changes must be made in the fan equipment due to the higher temperature and greater volume of the gases to be handled. That is, when air is heated it expands and in order to supply a given volume to the ash-pit at boiler room temperature it is necessary to draw through the fan a considerably larger volume at a higher temperature, thus requiring an increased speed of fan and more power. For average flue gas temperatures of 500 to 550 degrees the correction factors for maintaining a constant pressure difference between the ash-pit and furnace are approximately as follows: Air volume, 0.7; speed of fan, 1.4; power, 1.4.

For example, Table LIX. shows that a 4-foot fan at 370 r. p. m. will maintain a pressure of 1 inch water column, and deliver 10,000 cubic feet of air per minute with a power expenditure of 4.4 H. P. when used for forced draft.

TABLE LIX.

FANS FOR FORCED DRAFT. PADDLE-WHEEL TYPE.

Static pressure in inches of water	2½ feet		3 feet		3½ feet		4 feet		4½ feet		5 feet		5½ feet		6 feet		7 feet	
	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.
	4,200		5,800		7,800		10,000		12,400		15,200		18,200		21,400		28,800	
0.5	510	1.6	430	2.2	360	3.0	320	3.9	290	4.8	250	5.9	230	7.0	210	8.3	180	11.2
1.0	600	1.9	480	2.6	420	3.5	370	4.4	330	5.6	290	6.8	270	8.2	250	9.6	210	13.0
1.5	710	2.3	580	3.1	500	4.2	440	5.4	400	6.7	350	8.2	330	9.8	290	11.5	250	15.5
2.0	780	2.3	640	3.4	550	4.6	480	5.9	430	7.3	380	8.9	360	10.6	320	12.5	280	16.8

TABLE LX.

"SIROCCO" SINGLE-INLET FANS FOR FORCED DRAFT, MULTI-VANE TYPE.

Static pressure in inches of water	2 feet		2½ feet		3 feet		3½ feet		4 feet		4½ feet		5 feet		5½ feet		6 feet	
	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.	r.p.m.	h.p.
	6,330		9,900		14,280		19,400		25,380		32,000		39,600		48,000		57,000	
0.5	390	1.5	312	2.3	260	3.3	223	4.4	195	5.8	173	7.3	156	9.0	142	10.8	130	12.8
1.0	474	2.2	380	3.3	315	4.7	277	6.4	237	8.4	211	10.5	190	13.2	173	13.7	158	16.6
1.5	554	2.9	443	4.5	369	6.3	316	8.7	277	11.3	246	14.2	221	17.9	202	21.3	185	23.2
2.0	626	3.8	500	5.8	416	8.3	357	11.3	313	14.7	278	18.4	250	22.7	228	27.6	209	32.6

If now the same fan be used to maintain the same pressure under induced draft the air volume will become $10,000 \times 0.7 = 7,000$ cubic feet; the required speed $370 \times 1.4 = 518$ r. p. m.; and the horse power $4.4 \times 1.4 = 6.2$. The use of these tables is best shown by a practical illustration.

Example.—A boiler plant burning 2,600 pounds of coal per hour requires a draft pressure of 1.5 inches of water. What size and speed of fan (paddle-wheel type), and what power of engine will be required for forced draft? What will be the requirements for induced draft?

$$\text{Volume of air per minute} = \frac{2,600 \times 230}{60} = 9,967 \text{ cubic feet.}$$

Assuming it to be 10,000, and referring to Table LIX., we find this calls for a 4-foot fan at 440 r. p. m., and requires 5.4 H. P., which increased 10 per cent is 6 H. P. If the same fan were used for an induced draft of the same intensity, the air volume would drop to $10,000 \times 0.7 = 7,000$ cubic feet per minute, hence, we must look for a volume of approximately $10,000 \div 0.7 = 14,300$ cubic feet in selecting a size of fan for economical operation. This corresponds most nearly with a 5-foot fan (Table LIX.), which, under the conditions of induced draft, will draw into the ash-pit $15,200 \times 0.7 = 10,640$ cubic feet of air per minute, at a speed of $350 \times 1.4 = 490$ r. p. m, and requires $8.2 \times 1.4 \times 1.10 = 12.6$ H. P. This gives a slightly larger capacity than is necessary, which may easily be cared for by reducing the speed of fan.

The data given in Tables LIX. and LX. will be found useful for determining the size of fan required when laying out a boiler plant, but it is usually best to have the speed and power fixed by the makers of the particular type of fan to be used.

Boiler Furnaces.

239. Special Requirements.—The furnace is an important detail of a boiler plant and should receive careful attention in order to adapt it to the kind of fuel to be burned. Many of the furnaces used in power work are either patented or employ special devices in their construction, hence it is impossible, in the available space, to treat of this subject except in the briefest manner. The object sought in all furnace design is the nearest possible approach

to perfect combustion without too expensive equipment. Whatever the type of furnace or grade of fuel there should be provision for an ample supply of air and sufficient space for mixing it with the gases of combustion. Bituminous coal requires more space for this purpose than anthracite owing to the greater amount of volatile matter which it contains.

The subject of smokeless combustion is fully treated in a number of standard books and has also been considered in numerous magazine articles during the past two or three years. The reader is therefore referred to these for detailed information regarding the most approved methods to be employed in the design of a furnace for any given set of conditions. In the case of water-tube boilers the type of furnace is often recommended by the makers, and detail drawings will usually be furnished upon request.

240. *Grates*.—The required grate area is given by the formula

$$G = \frac{H. P. \times 34.5}{C \times E}$$

in which

G = grate area, in square feet.

H. P. = horse power of boiler.

C = rate of combustion, (from Table LIV.).

E = rate of evaporation, (from Table LV.).

Another formula is $G = \frac{H. P. \times W}{C}$

in which

W = pounds of coal per horse power per hour, (from Table LVI.), the other factors being the same as above.

Both stationary and shaking grates are employed in this class of work, although the former are confined principally to plants of small size. The principal disadvantages of stationary grates are the difficulty experienced in cleaning, especially with coal that clinkers badly, and the necessity of opening the furnace doors at this time which tends to check the fire and reduce the steam pressure. The size of a grate of this type is also limited because it is not possible to clean the rear portion properly by hand if over 6 feet in length when burning bituminous coal. Shaking

grates, on the other hand, may be cleaned without opening the furnace doors, and also require less time and manual labor, all of which is in the line of economy and more efficient operation.

Hand-fired furnaces require more skill on the part of the fireman and are more likely to produce smoke on account of intermittent firing and the larger quantity of fuel introduced at one time.

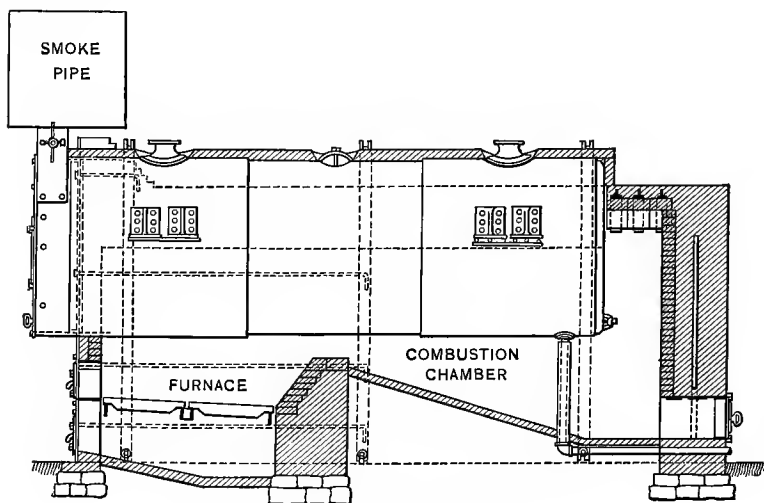


Fig. 99. Furnace for Return Tubular Boiler.

241. *Form of Furnace.*—The usual arrangement of furnace and combustion chamber for a return tubular boiler is shown in Fig. 99.

The height from grate to the under side of the shell should be from 24 to 30 inches for anthracite coal and from 36 to 48 inches for bituminous, depending upon the diameter of the boiler. Another form of furnace designed for smokeless combustion is illustrated in Figs. 100 and 101. In this case the grate is divided into two parts by a wall as shown in Fig. 101. These two fire-boxes connect with a common combustion chamber by way of a brick arch at the rear of the bridge wall. The advantage of this arrangement is that alternate firing may be employed, thus always having one bright fire and a continuous flow of hot gases

passing under an incandescent arch for burning the freshly distilled hydrocarbons from the new fire. This serves to illustrate the general principles involved in a considerable number of so-called smokeless furnaces.

A typical furnace for burning small sizes of anthracite coal under a water-tube boiler is shown in Fig. 102.

The Dutch oven furnace (Fig. 103) is a popular form for the burning of bituminous coal. The grate in this case is contained in a special chamber extending in front of the boiler, thus giving a large space directly beneath the tubes for burning the hydrocarbons as they are given off from the freshly fired coal. Various

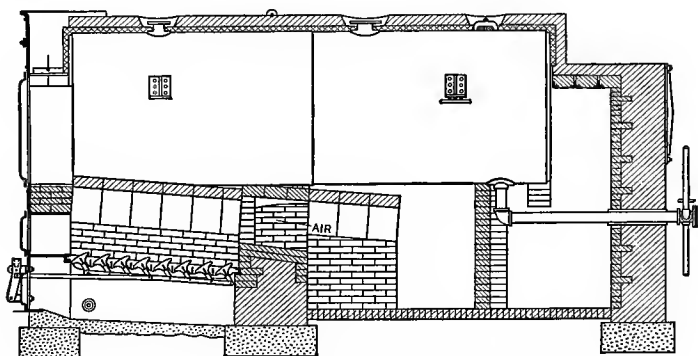


Fig. 100. Furnace Designed for Smokeless Combustion.

arrangements of baffle walls and arches serve to produce a thorough mixing of the air and gases of combustion.

242. Mechanical Stokers.—These are generally employed in plants of 1,500 H. P. or over, and have also been found desirable in many cases where the boiler capacity did not exceed 200 or 300 H. P.

The grade of fuel to be used, cost of skilled labor, smoke regulations, etc., are factors which govern their use to a large extent. One efficient fireman can attend to about 200 H. P. of hand-fired boilers under average conditions, while with mechanical stokers and overhead coal bins, with chutes, he can easily care for a plant of 1,000 to 2,000 H. P., depending upon the grade of fuel and type of equipment. The chief advantages of this method over

hand firing are a uniform supply of coal without opening the feed doors, greater accuracy in proportioning the air supply to the fuel, self-cleaning grates, and lower cost of labor.

Stokers are of two general classes, known as the "over-fed," and "under-fed." The first of these includes the inclined or step-grate stoker, one form of which is shown in Fig. 104. Stokers of this type are adapted to practically all the grades of fuel commonly used in the East, and in general, may be operated in connection

with a natural draft if desired, although some make use of a steam jet under heavy loads.

The chain grate stoker (Fig. 105), is widely used in the West, and in other localities for burning the smaller sizes of bituminous coal.

Under-fed stokers, as the name implies, supply the fuel from below, thus conducting the volatile gases through the incandescent fuel bed, where they are raised to a high temperature before mixing with the air in the furnace.

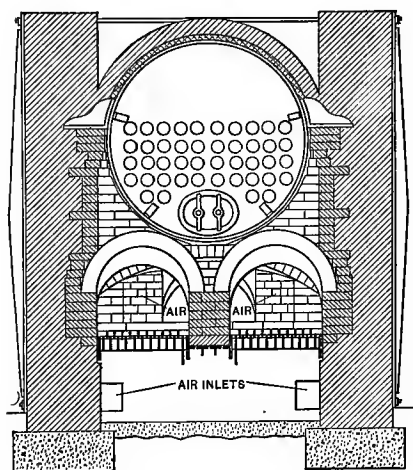


Fig. 101. Transverse Section through Boiler Shown in Fig. 100.

Stokers of this type require a forced draft and are able to handle heavy overloads with but little if any smoke. They may be used for all kinds of bituminous coal, and owing to the fact that forced draft is employed, are especially adapted to the burning of the cheaper grades.

An under-fed stoker of the gravity type is shown in Fig. 106, and is sufficiently explained by the notes on the drawing.

The method of installing this stoker in connection with a water-tube boiler is illustrated in Fig. 107. In general, an efficient stoker should be simple in construction, in order to reduce the danger of a breakdown to a minimum. While automatic to a consider-

able extent, they, like all other mechanical devices, require a certain amount of care and adjustment to obtain the best results. The steam required for maintaining a forced draft and operating a mechanical stoker will vary from 2 to 5 per cent of that generated by the boiler, depending upon the conditions of operation.

Special Equipment.

Under this heading may be included superheaters and economizers, both of which are used to some extent in isolated plants.

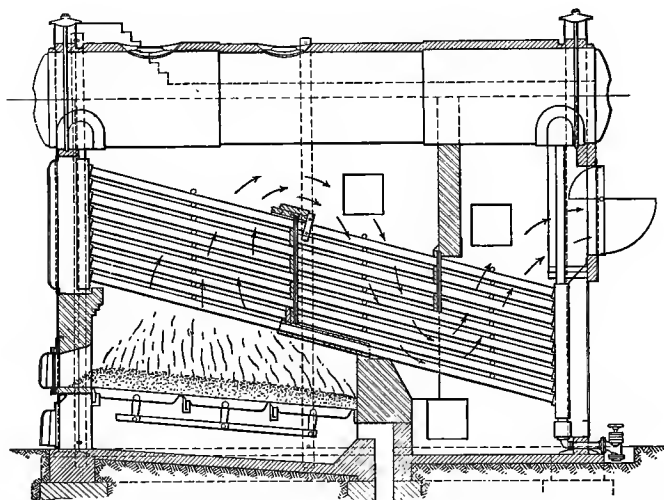


Fig. 102. Furnace for Burning Anthracite Coal Under a Water-Tube Boiler.

243. *Superheaters.*—These are of two general types, the *separately fired*, and that contained in the regular boiler setting, known as the *indirectly fired*. While the former has certain advantages in the way of independent regulation, the indirect type is more frequently used in isolated plants owing to its lower cost and greater compactness, the latter being a matter of considerable importance in basement plants. Fig. 108 shows a typical form of superheater, placed in the smoke passage above the boiler, and also illustrates a common arrangement for a system of induced draft.

The required amount of superheating surface depends upon various conditions, among which the more important are the location in relation to the furnace, the velocity of both the hot gases and steam, and also the degree of superheat required. For average conditions, with steam at 160 to 170 pounds gauge pressure, and 100 to 150 degrees superheat, the following amount of surface will be required per horse power.

TABLE LXI.

SUPERHEATING SURFACE.

Location of superheating surface.	Square feet of surface per boiler horse power.
In the furnace.	0.20 to 0.25
At the end of the first pass.	2 " 3
At the end of the heating surface.	3 " 4

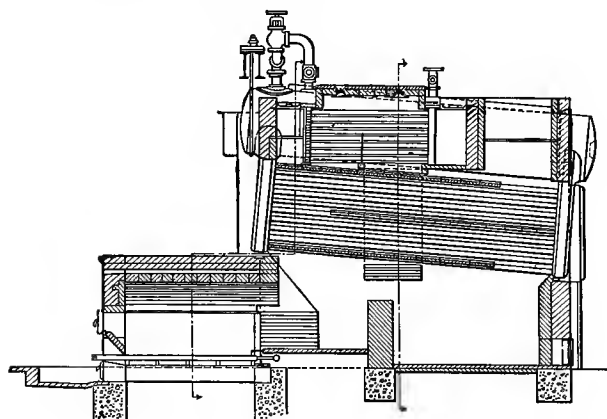


Fig. 103. Dutch Oven Furnace.

The gain in economy by the use of superheated steam is considered in the following chapter.

244. *Economizers*.—While an economizer is usually considered as forming a part of the feed-water equipment it is in reality an extension of the boiler, being made up of auxiliary heating surface so arranged as to extract heat from the waste gases which

would otherwise pass to the chimney. It is not adapted to the generation of steam, but serves to heat the water to a compara-

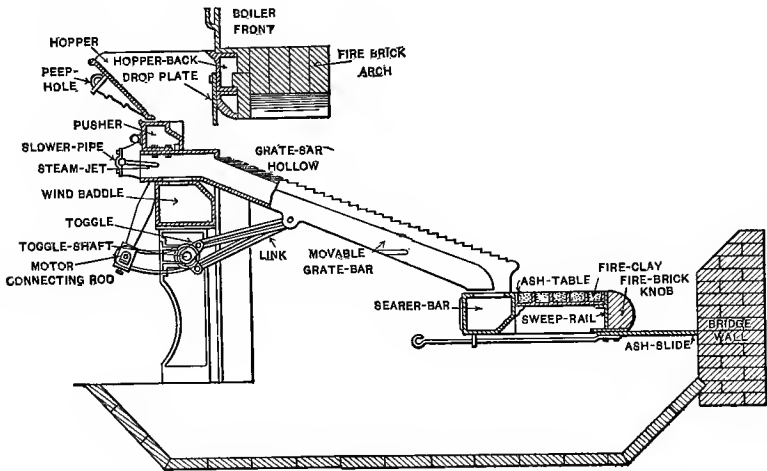


Fig. 104. Inclined Over-Fed Stoker.

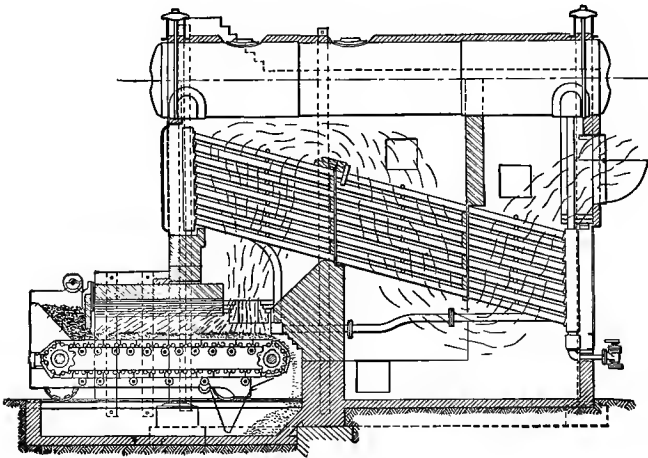


Fig. 105. Chain Grate Stoker.

tively high temperature, thus relieving the heating surface of the boiler and making available a greater capacity for evaporation.

In isolated plant work economizers are more frequently used in factories and the power houses of institutions than in the basement plants of city buildings, although special designs of compact form are made for this particular class of work. Typical economizer installations are shown in Figs. 108 and 121.

Economizers may be employed for increasing the capacity of boilers already installed or for purposes of fuel economy by utilizing the heat from the waste gases of combustion. They also

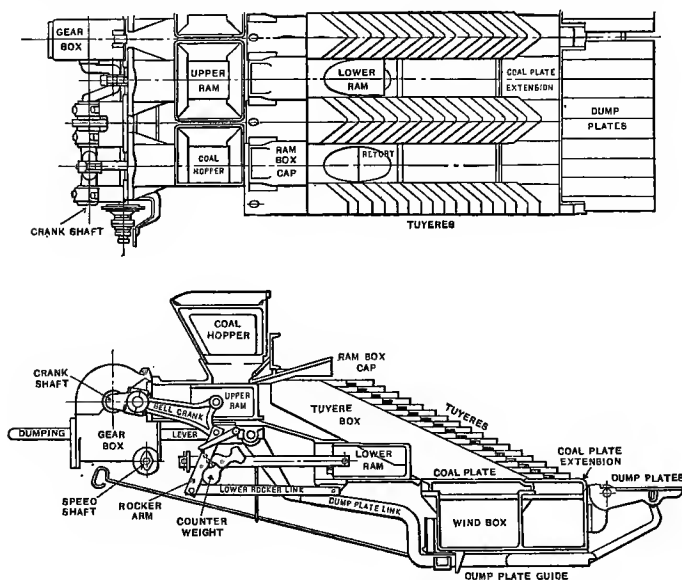


Fig. 106. Gravity Under-Fed Stoker.

add to the life of the boilers by furnishing feed water at a high temperature and provide a large storage of hot water for offsetting to some extent sudden calls for steam.

Where space is limited it is usually more economical, in case of a new plant, to place the entire heating surface within the boilers, on account of its greater efficiency and the smaller amount required. When there is ample exhaust steam for heating the feed water the only advantage in using an economizer is in the higher

feed temperature obtainable, and this may be offset by the cost of installation and operation.

A series of tests in which the average temperature of the gases entering the economizer was 504 degrees, and the total rise in water temperature 117 degrees, showed an average saving in fuel of 13.4 per cent. Another series of tests with an average rise in

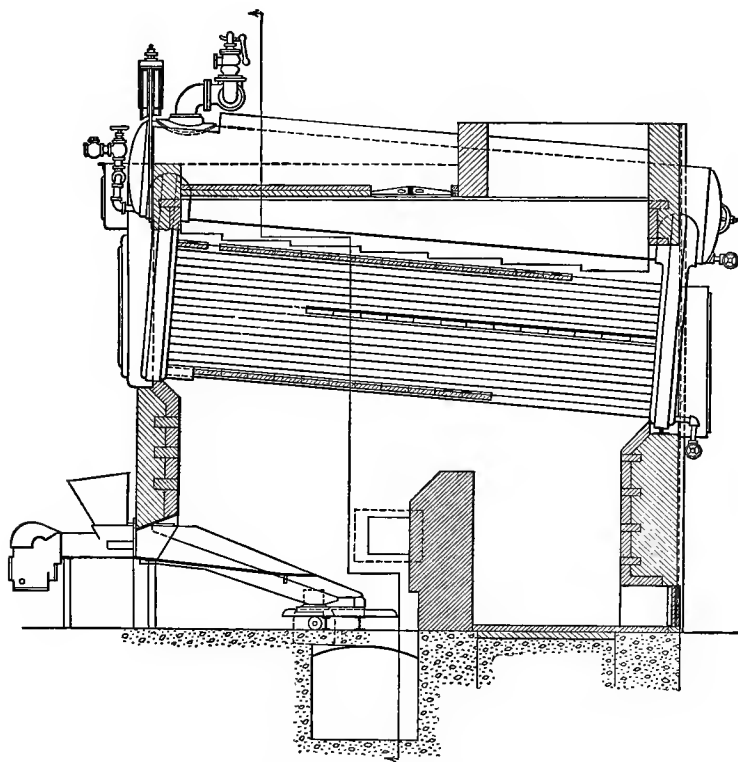


Fig. 107. Under-Fed Stoker Installed Under a Water-Tube Boiler.

feed temperature of 140 degrees showed a saving in fuel of 13.8 per cent. The average results of the two series give an increase in economy of 1 per cent for each 9.4 degrees rise in feed temperature, which corresponds well with ordinary practice, although better results may be obtained in special cases. Under average conditions the reduction in flue gas temperature will run from

220 to 260 degrees, and the rise in feed temperature will be about one-half that.

In order to determine the advisability of installing an economizer in any given case the question of space should be considered, also the quantity and use of exhaust steam and the tem-

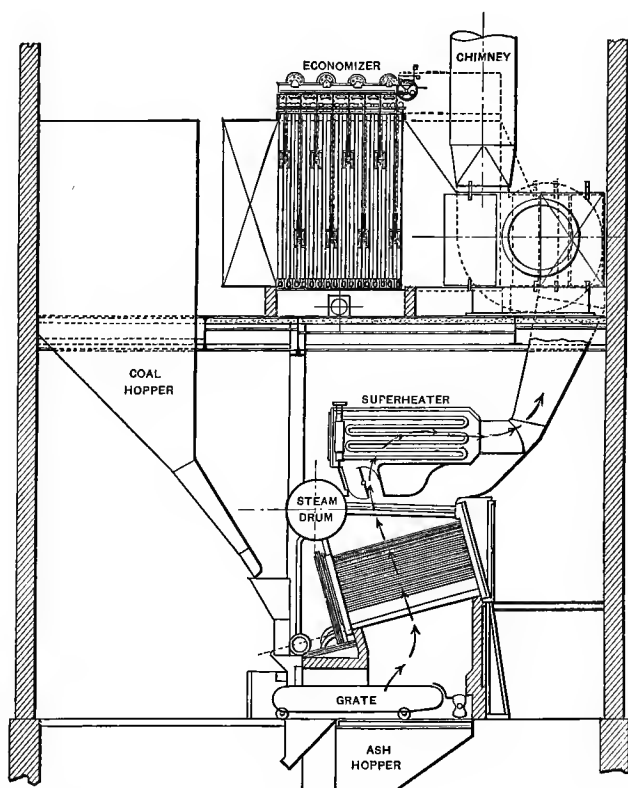


Fig. 108. Diagram Showing Superheater and Economizer in a Typical Plant.

perature of the return water or condensation coming back from the heating system. The higher the temperature of the return the less will be the gain by the use of an economizer. By means of the above data the approximate saving in fuel may be computed, and by comparing this with the cost of operating an econ-

omizer, including interest, taxes, repairs, depreciation, etc., the net results may be easily determined.

The amount of surface required will vary according to local conditions but commonly runs from 4 to 5 square feet per boiler horse power. The necessary space and best arrangement in any given case may be obtained from the manufacturers.

Boiler Settings.

245. Settings for Water-Tube Boilers.—Water-tube boilers are usually installed by the makers or their selling agents; otherwise,

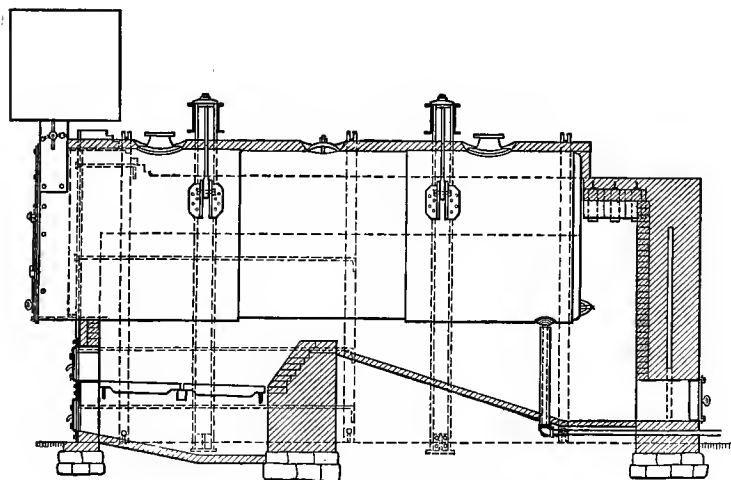


Fig. 109. Setting for a Return Tubular Boiler.

detail drawings are furnished for both foundations and settings. Boilers of this type vary so widely in their form and construction that no general method of setting can be followed.

246. Settings for Fire-Tube Boilers.—Settings for return tubular boilers are more uniform and have been considered in detail in a previous volume. Drawings may be obtained from the manufacturers or from any reliable boiler inspection and insurance company.

Two standard forms of settings for boilers of this type are shown in Figs. 99 and 109. The foundation is usually of concrete, while the setting itself should be of hand-burned brick. The furnace, bridge wall and combustion chamber are lined with fire-brick.

The older form of setting, in which the boiler is supported by brackets resting upon the side walls, is shown in Fig. 99. The principal objection to this is the liability of cracks forming in the setting due to the expansion and contraction of the shell. A better arrangement for power work is to suspend the boiler from overhead girders supported by columns, thus taking the entire

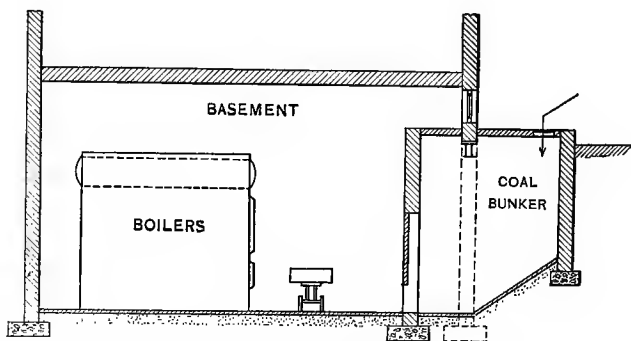


Fig. 110. Simple Arrangement for the Storage of Coal.

weight from the setting, (Fig. 109). Sometimes a combination is used, the front of the boiler resting upon brackets while the rear portion is supported by lugs and hangers.

With the arrangement shown in Fig. 99 the outer walls are usually made 30 inches in thickness, consisting of a 16-inch inner wall, a 2-inch air space, and a 12-inch outer wall, the latter sometimes being reduced to 8 inches in case of the smaller sizes. When the boilers are suspended, lighter walls may be used as they carry only their own weight.

Two 8-inch walls, with a 2-inch air space between, gives sufficient thickness for boilers up to 54 inches in diameter. For larger sizes the inner wall should be increased to 12 inches.

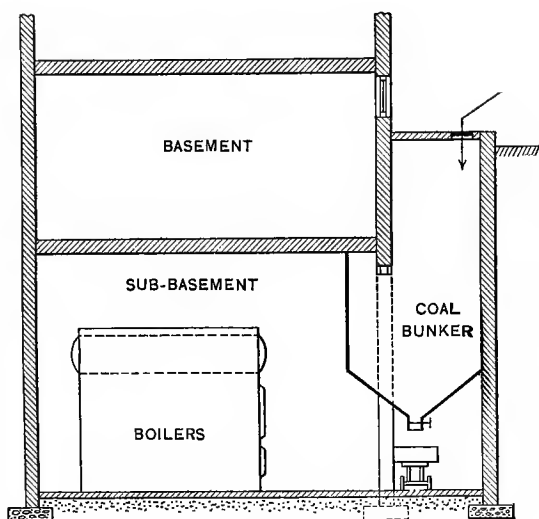


Fig. 111. Coal Bunker in Sub-Basement.

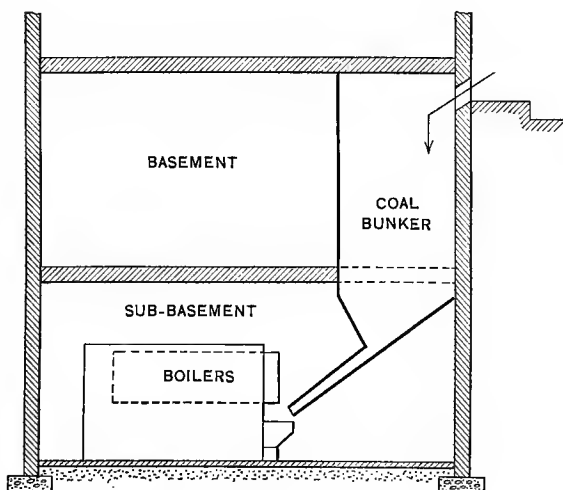


Fig. 112. Coal Bunker with Chute to Stoker.

The Storage and Handling of Coal.

247. *Arrangements in City Buildings.*—The storage and handling of coal and the removal of ashes are important matters in the design of any steam plant. In city buildings conditions vary so widely that no uniform method can be followed in detail. One of the simplest methods, where the boiler room is only one story below the street level, is shown in Fig. 110. The coal in this case is brought in wagons or trucks and dumped through scuttles

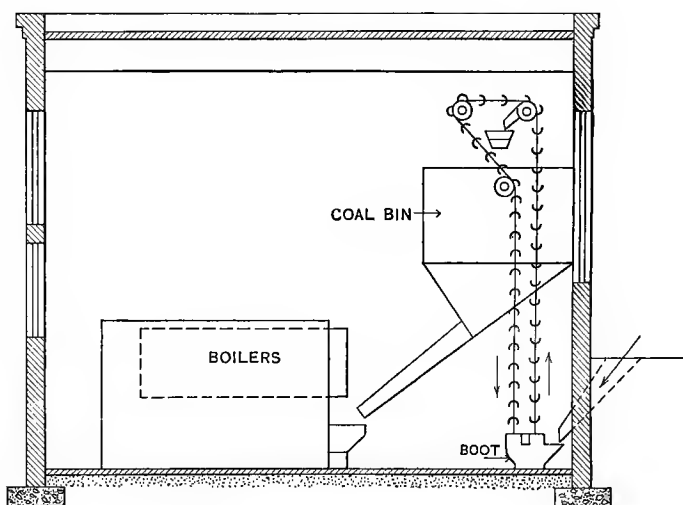


Fig. 113. Overhead Coal Bin with Elevator.

in the sidewalk into bins below, from which it is shoveled into cars and delivered in front of the boilers, or fired directly from the cars.

When the boilers are located in a sub-basement, the arrangement shown in Fig. 111 may be used, thus doing away with one handling. Here the bins are elevated sufficiently to allow of the coal being delivered to the cars or barrows by gravity, and in some cases conditions are such that the chutes may be made to discharge directly in front of the furnaces, as in Fig. 112.

248. *Coal and Ash Conveyors.*—In the power plants of factories and institutions, where more space is available, conveyors and overhead bins are often provided.

A simple lay-out for a plant of this kind is shown in Fig. 113. In this case the coal is delivered as in Fig. 110, but into a boot or hopper connected with a bucket conveyor which discharges

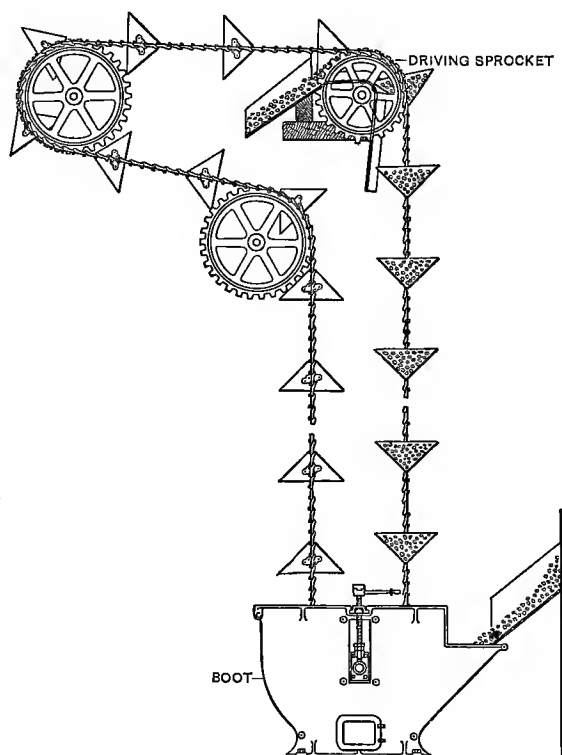


Fig. 114. Detail of Coal Elevator.

it into overhead bins from which it may be fed into the stoker hoppers or dumped in piles in front of the furnaces as desired.

The general form of the buckets and the pulley arrangement are shown in diagram in Fig. 114. Conveyors of this type are usually driven by an electric motor (not shown in the cut), located at the top and geared to the sprocket near the delivery chute as in-

dicated. Each boiler is commonly provided with a separate bin with sloping sides and the conveyor furnished with a tripping device which delivers the coal into each in turn as may be desired. Equipment of this kind may also be adapted to do a part of the work, at least, in city buildings where the boiler plants are of large size. More elaborate outfits are shown in

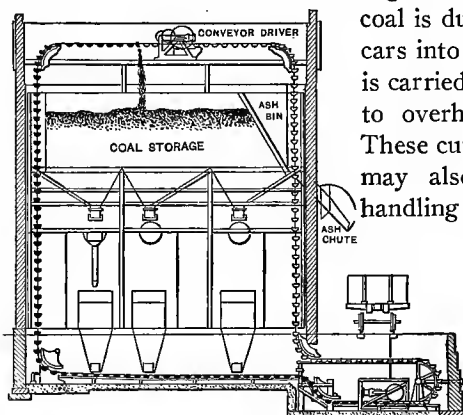


Fig. 115. Coal Conveyor and Storage Bins for Large Plant.

Figs. 115 and 116. Here the coal is dumped directly from the cars into crushers, from which it is carried by means of conveyors to overhead bins as indicated. These cuts show how a conveyor may also be utilized for the handling of ashes as well as coal.

Spouts are run from the ash-pits either directly to the conveyor or arranged to discharge into cars, the contents of which are easily dumped into the conveyor buckets

and carried to an overhead bin. From here the ashes are delivered into cars or wagons through a special chute as shown.

"Vacuum" ash-conveyors are employed in some plants, the equipment being similar to that used in wood-working mills for the removal of shavings and saw-dust. The ashes are usually raked by hand from the ash-pits into floor openings connecting with the suction pipe and discharged into specially constructed bins. In other cases branch suction pipes connect directly with the ash-hoppers beneath the boilers. These cuts illustrate only a few of the methods employed, and the reader is recommended to examine catalogues relating to this kind of equipment for more detailed information upon the subject.

249. Storage Capacity.—The storage capacity for coal in city buildings is usually limited to a three or four days' supply and even in the case of factories and institutions it is commonly necessary to provide outside storage bunkers from which the coal may be easily conveyed to the power house.

In designing coal bunkers, from 40 to 45 cubic feet of storage space per ton should be allowed for bituminous coal and about 35 cubic feet for anthracite.

Ventilation of Boiler and Engine Rooms.

250. Buildings With Monitor Roofs.—The ventilation of an isolated power house with monitor roof and outside windows is a comparatively simple matter and may be accomplished by providing swing sashes, so arranged as to produce an upward movement of air from the lower windows through the monitor. Ven-

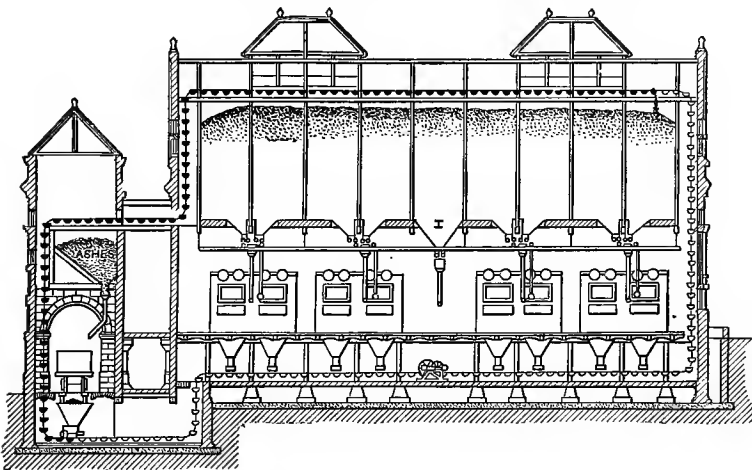


Fig. 116. Typical Lay-Out for the Mechanical Handling of Coal and Ashes.

tilators or other roof openings should never be placed over engines or generators owing to the danger of leakage. Sufficient circulation of air can easily be obtained through swing sashes located on opposite sides of the engine room.

251. Sub-Basement Rooms.—When the plant is located in the sub-basement of a city building special equipment must be provided for supplying fresh cool air and for removing the heated air from both boiler and engine rooms.

Cool air can only be supplied by means of a special fan and should be delivered through down-takes in front of the boilers at

a height of 7 or 8 feet from the floor. Care should be taken to direct the currents of entering air in such a manner as not to blow directly upon those working below. The hot air may usually

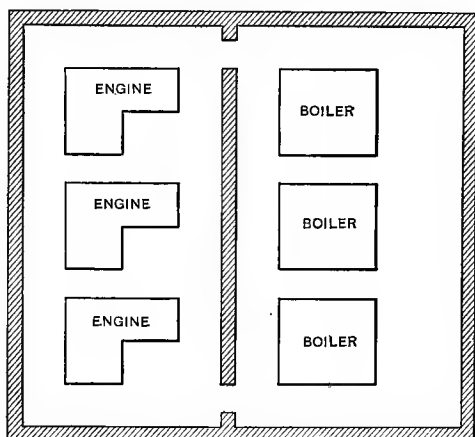


Fig. 117. Back to Back Arrangement of Boilers and Engines.

be removed through a vent shaft leading to the top of the building, the height of the flue and the high temperature of the air producing sufficient draft in most cases without the use of fans. Engine rooms may be treated in practically the same manner.

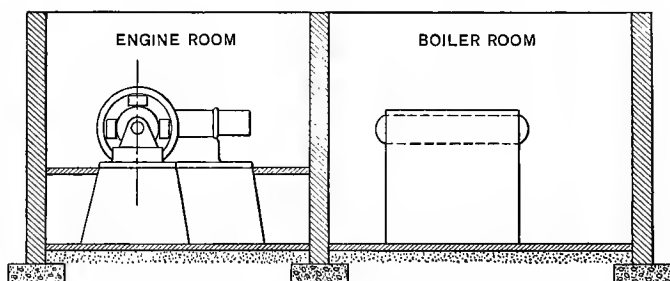


Fig. 118. Transverse Section through Arrangement in Fig. 117.

The air volume will depend somewhat upon local conditions, but should, in general, amount to 10 or 12 changes per hour.

When the power plant is located directly below occupied rooms ample insulation should be provided at the ceiling of the engine and boiler rooms. Sometimes a space 18 to 24 inches in depth is constructed and the air for one or more of the ventilating fans drawn through it, thus utilizing the waste heat. This, however, can only be done in the winter when heat is needed and other means must be provided for summer use.

Plant Arrangement.

252. General Considerations.—The general arrangement of the power equipment will depend largely upon local conditions. In the case of an independent or isolated power house certain standard arrangements of boilers and engines may often be employed, either wholly or in part. In basement plants, however, the available space varies so greatly in form and amount that

no general system can be followed and each individual case must be worked out according to the special conditions involved.

253. Typical Arrangements.—When there is sufficient leeway a modification of either the “back-to-back” or “end-to-end” system is commonly employed. The first of these, shown in diagram in Fig. 117, is the more compact, and therefore preferable under ordinary conditions. When possible the engine room floor should be at a higher elevation, as shown in Figs. 118 and 121, in order to provide space for condensing apparatus and exhaust piping. The “end-to-end” system is illustrated in Fig. 119.

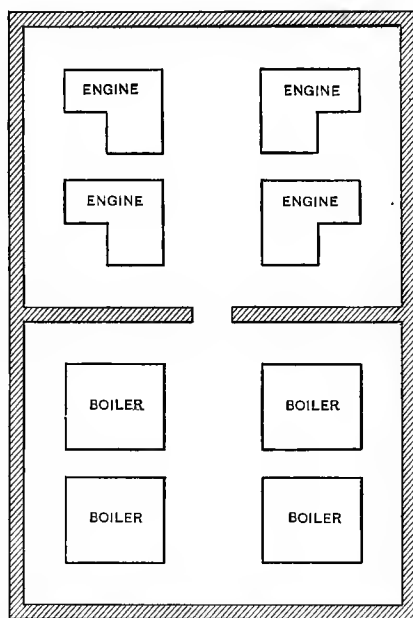


Fig. 119. End to End Arrangement of Boiler and Engine Rooms.

In both cases all of the boilers are fired from a single aisle, which is a desirable feature. A section through a typical boiler room of large size, is shown in Fig. 120, and illustrates the general arrangement of the more important parts of the equipment.

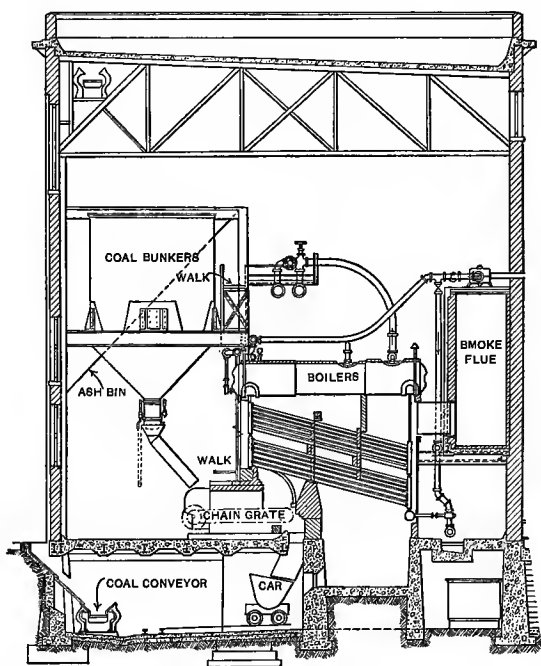


Fig. 120. Section through a Boiler Room of Large Size.

Fig. 121 shows a similar plant, including the engine room and condenser equipment. These illustrations give only a few suggestions for the design of new plants and should be supplemented by a study of descriptive articles in late issues of the standard engineering magazines and by an inspection of recent installations similar to the one under consideration.

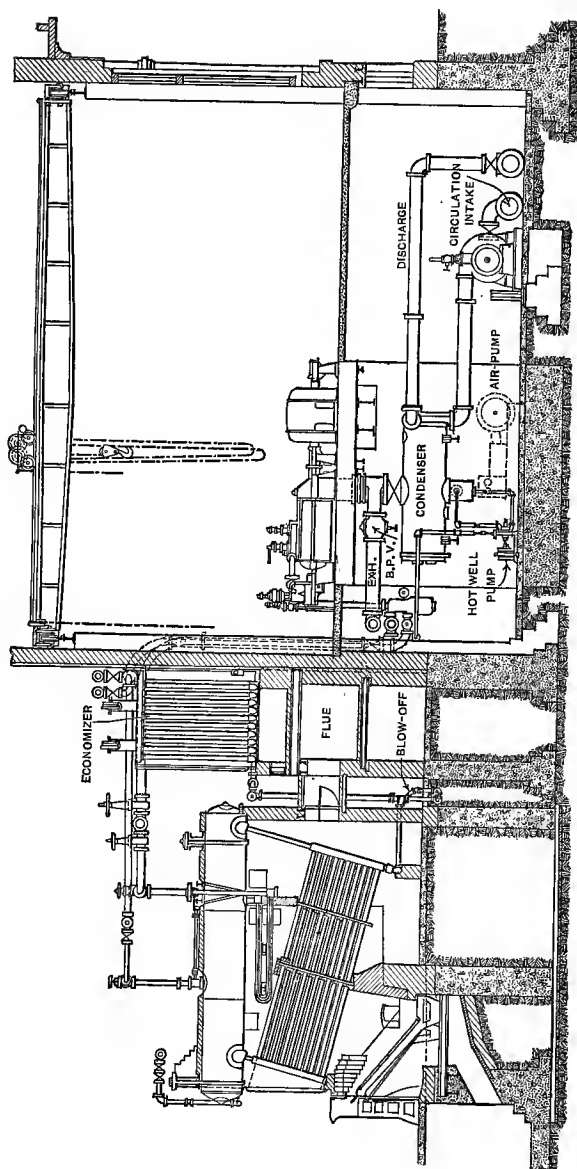


Fig. 121. Section through Boiler and Engine Rooms in a Typical Plant.

CHAPTER VII.

STEAM ENGINES AND TURBINES.

Selection of an Engine.

254. *Points to Be Considered.*—In selecting a type of engine for a given plant the matter of economy should be considered from different standpoints.

First the importance of a low water-rate should be taken into account. If the engine is for a plant where all, or a large proportion of the exhaust steam can be utilized for heating, or in manufacturing processes, there is evidently no advantage in adding to the fixed charges by installing a high priced machine for the sake of steam economy. On the other hand, if a considerable quantity of exhaust is wasted at different seasons of the year, a higher priced engine may prove more economical, the amount of saving increasing with the proportion of exhaust discharged outboard. In this connection a study should be made of the length of time during which the supply and demand for exhaust steam overlap each other. In other words, the amount of exhaust which may be *utilized* is the important point, rather than the amounts supplied and required when considered separately.

Problems relating to the practical application of the above are worked out in detail in Chapters X. and XII.

Other matters to be considered in the selection of an engine are available space, simplicity of construction, amount of power required and the purpose for which it is to be used.

Types of Engines.

255. *Those Adapted to Isolated Plants.*—The types of engines commonly employed in isolated plants may be included under the general heads of high and medium speed, with the addition of low-speed engines in the case of manufacturing plants. These general types are sub-divided into single and multi-valve engines, simple, compound, condensing and non-condensing, according to their design and the conditions under which they are operated.

256. Simple High-Speed Engines.—These are of two general forms, the single-valve and the multi-valve. The first of these, shown in Fig. 122, is commonly employed for sizes up to about 200 I. H. P. where floor space is limited, and especially when steam economy is not of the first importance. Engines of this type and size are seldom operated condensing because any gain in economy is usually offset by the cost of the condensing equipment. The speed depends upon the size, running from about 200 to 300 r. p. m. under average conditions. The steam consumption with non-condensing

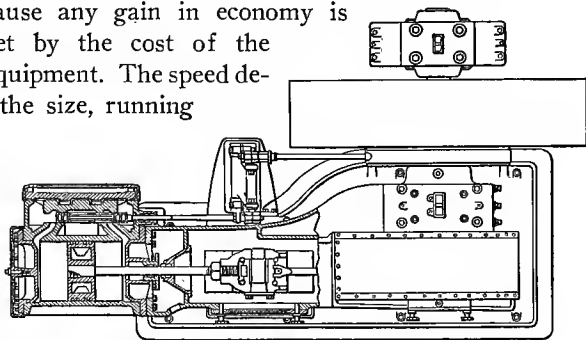


Fig. 122. Section through a Single-Valve High-Speed Engine.

engines of this type will average from 30 to 32 pounds per I. H. P. per hour with the form commonly employed for driving electric generators.

Small slide-valve engines used in connection with fans, laundry work, etc., often require from 40 to 50 pounds of steam per I. H. P. The principal defects of the single-valve engine are the small economical load range and the large clearance. These are both overcome, to a considerable extent, in the multi-valve engine (see Fig. 123), which results in lowering the steam consumption to an average of 28 to 30 pounds per I. H. P. In general, four valves give a better economy than a less number as the different events of the stroke are entirely independent and a much better adjustment is possible.

The principal disadvantages are the increased cost, the greater number of parts, and the more careful attention required. Engines of this type are built in sizes up to about 1,000 I. H. P.

257. Simple Medium-Speed Engines.—Between the high and low-speed multi-valve engines are those of medium speed, running from 100 to 175 r. p. m. They are equipped with non-releasing valves, commonly of the Corliss type, and have an average steam

consumption of 26 to 28 pounds per I. H. P. per hour. Medium-speed engines are built in sizes up to 1,500 I. H. P. or more and are especially adapted to isolated plants when greater steam economy is desired or when high-speed engines are likely to produce vibration.

258. Simple Low-Speed Engine.—This type of engine is confined almost exclusively to manufacturing plants on account of the floor space required, and is built in sizes up to about 3,000 I. H. P. When equipped with a releasing gear the speed is limited to about 100 r. p. m. in the smaller sizes and to 75 r. p. m. in larger ones.

Owing to the small clearance space and the sharp cut-off possible with this form of valve a steam economy of 24 to 26 pounds is easily obtained with machines of medium size.

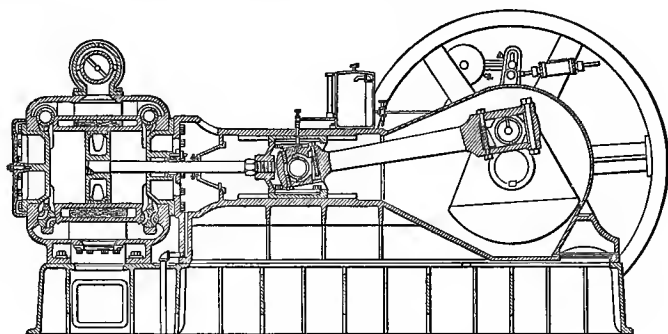


Fig 123. Section through a Four-Valve Engine.

259. Compound Engines.—When the conditions are such that a high steam economy is required, compound engines may be used. These are of the same general type as the simple engines above described, with the exception that expansion takes place in two cylinders instead of one, thus reducing the temperature range in a single cylinder and so diminishing condensation.

The types most commonly employed in isolated plants, where floor space is limited, are the tandem and angle engines, shown in Figs. 124 and 125 respectively.

Cross-compound engines, which are made up of two complete machines, except for the main shaft and fly-wheel, are confined largely to central plants on account of the floor space required.

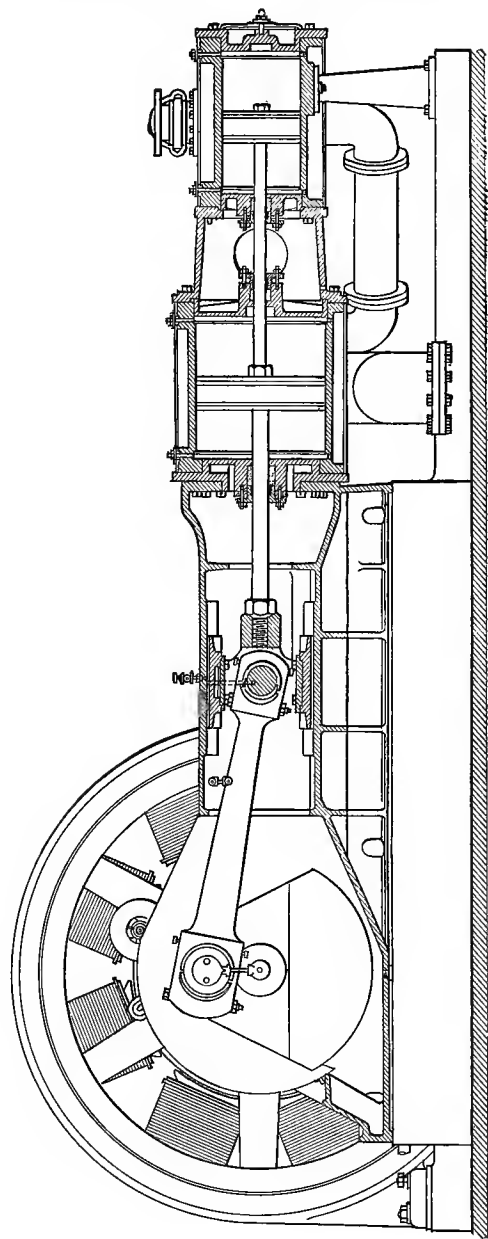


Fig. 124. Section through a Tandem Compound Engine.

It has been stated that the gain in economy by compounding is due to a reduction in cylinder condensation. While this is the immediate effect, the principal gain comes from the ability to use higher initial pressures and larger ratios of expansion without excessive losses from condensation. Under suitable conditions a saving of 15 to 20 per cent at full load may be made by compounding in case of a non-condensing engine and 25 to 30 per cent when condensing.

The disadvantages of a compound engine are the higher first cost, greater number of parts to care for, increased space, and greater radiation losses.

The best ratio between the volumes of the high and low-pressure cylinders will depend somewhat upon the speed and initial steam pressure. In the case of high-speed single valve engines with steam pressures of 100 to 150 pounds the volume of the low-pressure cylinder is commonly made from $2\frac{1}{2}$ to 3 times that of the high-pressure cylinder, while with low-speed condensing engines with steam pressures of 125 to 175 pounds, the ratio commonly runs from 3 to 4.

Compound engines are especially adapted to cases where only a comparatively small part of the exhaust can be utilized or where the non-heating season extends through a considerable portion of the year.

Steam Engine Economy.

260. *Steam Tables.*—As problems in steam engine economy require frequent reference to steam tables, the following has been

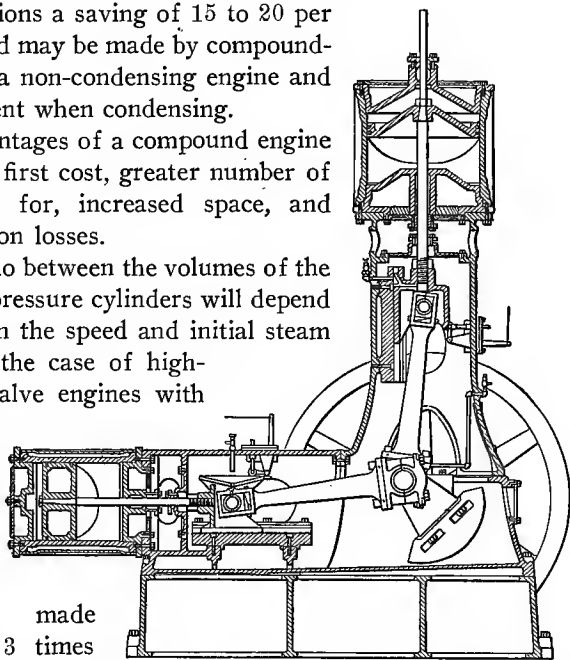


Fig. 125. Section through an Angle Compound Engine.

abridged from more complete tables for use in connection with the various examples which appear later in the text.

TABLE LXII.

THE PROPERTIES OF SATURATED STEAM.

(Based on Marks and Davis Steam Tables and Diagrams, by permission, Copyright 1909 by Longmans, Green & Company.)

Pressure in pounds per square inch.	Temperature, in degrees Fahr.	Heat in the liquid above 32 degrees.	Latent heat of evaporation.	Total heat of evaporation.	Weight of a cubic foot of steam, in pounds.	Volume of a pound of steam, in cubic feet.
Absolute pressure						
1	101.8	69.8	1034.6	1104.4	0.0030	333.0
2	126.2	94.0	1021.0	1115.0	0.0058	173.5
3	141.5	109.4	1012.3	1121.6	0.0085	118.5
4	153.0	120.9	1005.7	1126.5	0.0111	90.5
5	162.3	130.1	1000.3	1130.5	0.0136	73.33
6	170.1	137.9	995.8	1133.7	0.0162	61.89
7	176.9	144.7	991.8	1136.5	0.0187	53.56
8	182.9	150.8	988.2	1139.0	0.0212	47.27
9	188.3	156.2	985.0	1141.1	0.0236	42.36
10	193.2	161.1	982.0	1143.1	0.0261	38.38
11	197.8	165.7	979.2	1144.9	0.0285	35.10
12	202.0	169.9	976.6	1146.5	0.0309	32.36
13	205.9	173.8	974.2	1148.0	0.0333	30.03
14	209.6	177.5	971.9	1149.4	0.0357	28.02
Gauge pressure						
0	212.0	180.0	970.4	1150.4	0.0373	26.79
1	216.3	184.4	967.6	1152.0	0.0404	24.79
2	219.4	187.5	965.6	1153.1	0.0428	23.38
3	222.4	190.5	963.7	1154.2	0.0451	22.16
4	225.2	193.4	961.8	1155.2	0.0475	21.07
5	228.0	196.1	960.0	1156.2	0.0498	20.08
6	230.6	198.8	958.3	1157.1	0.0521	19.18
7	233.1	201.3	956.7	1158.0	0.0545	18.37
8	235.5	203.8	955.1	1158.8	0.0568	17.62
9	237.8	206.1	953.5	1159.6	0.0591	16.93
10	240.1	208.4	952.0	1160.4	0.0614	16.30
15	250.3	218.8	945.1	1163.0	0.0728	13.74
20	259.3	227.9	938.9	1166.8	0.0841	11.89
25	267.3	236.1	933.3	1169.4	0.0953	10.49
30	274.5	243.4	928.2	1171.6	0.1065	9.39
35	281.0	250.1	923.5	1173.6	0.1175	8.51
40	287.1	256.3	919.0	1175.4	0.1285	7.78
45	292.7	262.1	914.9	1177.0	0.1394	7.17
50	298.0	267.5	911.0	1178.5	0.1503	6.65
55	302.9	272.6	907.2	1179.8	0.1612	6.20
60	307.6	277.4	903.7	1181.1	0.1721	5.81
65	312.0	282.0	900.3	1182.3	0.1829	5.47
70	316.3	286.3	897.1	1183.4	0.1937	5.16
75	320.3	290.5	893.9	1184.4	0.2044	4.89
80	324.1	294.5	890.9	1185.4	0.2151	4.65
85	327.8	298.3	888.0	1186.3	0.2258	4.43
90	331.4	302.0	885.2	1187.2	0.2365	4.23
95	334.8	305.5	882.5	1188.0	0.2472	4.05
100	338.1	309.0	879.8	1188.8	0.2577	3.88
110	344.4	315.5	874.7	1190.3	0.2791	3.58
120	350.3	321.7	869.9	1191.6	0.3002	3.33
130	355.8	327.4	865.4	1192.8	0.3213	3.11
140	361.0	332.9	861.0	1194.0	0.3425	2.92
150	366.0	338.2	856.8	1195.0	0.3633	2.75
160	370.8	343.2	852.7	1195.9	0.3843	2.60
170	375.4	348.0	848.8	1196.8	0.4052	2.47
180	379.8	352.7	845.0	1197.7	0.4262	2.35
190	384.0	357.1	841.4	1198.5	0.4470	2.24
200	388.0	361.4	837.9	1199.2	0.4680	2.14

NOTE—Gauge pressures were obtained by subtracting 15 from the corresponding absolute pressures in the original table, which gives results sufficiently accurate for the problems solved in the present volume.

261. *Thermal Efficiency.*—The thermal efficiency of an engine or turbine is the ratio of the heat converted into useful work to that supplied, measured above the heat of the liquid at exhaust pressure. This is equal to

$$\frac{2,546}{W(H-q)}$$

in which

2,546 = the heat equivalent of 1 horse power, in B. T. U. per hour.

W = pounds of steam required per developed horsepower (D. H. P.), per hour.

H = heat in 1 pound of steam at initial pressure = (latent heat $\times$ quality of steam) + heat in liquid.

q = heat in liquid at exhaust pressure.

Example.—What is the thermal efficiency of an engine operating under the following conditions?

Initial steam pressure.....120 pounds guage.

Quality of steam..... 98 per cent.

Exhaust pressure..... 1 pound guage.

Pounds of steam per D. H. P. per hour 30.

From Table LXII. we have

Latent heat at 120 pounds pressure.....870 B. T. U.

Heat in liquid at 120 pounds pressure.....322 B. T. U.

Heat in liquid at 1 pound pressure.....184 B. T. U.

Hence,

$$H = (870 \times 0.98) + 322 = 1,175 \text{ B. T. U.}$$

Substituting the above values in the formula gives a thermal efficiency of

$$\frac{2,546}{30(1,175 - 184)} = 0.086 \text{ or } 8.6 \text{ per cent.}$$

When W is given in pounds of steam per kilowatt-hour the formula becomes

$$\frac{3,413}{W(H-q)}$$

In case of superheated steam the value of H may be taken directly from a table giving the properties of superheated steam or

computed from the total heat of dry saturated steam at the same pressure, by means of the specific heat for the given conditions.

Table LXIII. gives the mean specific heat of superheated steam for different pressures and degrees of superheat.

TABLE LXIII.

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM.

Gauge pressure.	Mean specific heat of superheated steam for different degrees of superheat.					
	50 degrees.	70 degrees.	100 degrees.	130 degrees.	160 degrees.	200 degrees.
60	0.528	0.523	0.517	0.512	0.507	0.504
85	0.550	0.542	0.534	0.526	0.520	0.515
110	0.570	0.560	0.548	0.539	0.533	0.524
135	0.592	0.579	0.562	0.550	0.541	0.533
160	0.614	0.597	0.577	0.562	0.551	0.541
185	0.634	0.614	0.590	0.572	0.559	0.548
210	0.656	0.631	0.603	0.583	0.567	0.553

Example.—What is the value of H for steam at 160 pounds gauge pressure with 130 degrees superheat?

The total heat in 1 pound of dry saturated steam at 160 pounds pressure is 1,196 B. T. U. (from Table LXII.), therefore, the total heat, or value of H , for the superheated steam is $1,196 + (130 \times 0.562) = 1,269$ B. T. U.

262. Mechanical Efficiency.—The ratio of the developed or brake horse power (D. H. P.), to the indicated horse power (I. H. P.) of an engine is called the *mechanical efficiency*, and is

expressed by the fraction $\frac{\text{D. H. P.}}{\text{I. H. P.}}$

The difference between these two quantities represents the power consumed in friction and commonly varies from 5 to 20 per cent of the I. H. P. With the machines employed in isolated plants it usually varies from 5 to 15 per cent, depending upon the type of engine and adjustment of the bearings.

While the friction increases to some extent with the load, it is customary with good lubrication, to take the no-load friction when computing the mechanical efficiency.

263. Water-Rate.—The operating economy of a steam engine is commonly given in pounds of steam per indicated horse power per hour, although in electrical work it is often expressed in

pounds of steam per kilowatt per hour. In the case of steam turbines it is given either in brake horse power or the output is expressed in kilowatts, as the indicated horse power of a turbine cannot be obtained. When making a comparison of the water-rates of engines and turbines it is therefore necessary to reduce the results to a common basis. If the mechanical efficiency of an engine or class of engines is known the brake horse power may be found by the equation

$$\text{D. H. P.} = \text{I. H. P.} \times \text{mechanical efficiency.}$$

Example.—The water-rate of an engine is 27 pounds of steam per I. H. P. per hour and the mechanical efficiency is 0.9, what is the water-rate per D. H. P.?

$$27 \div 0.9 = 30 \text{ pounds.}$$

The kilowatt rating of a generating unit may be changed to a D. H. P. rating of the driving engine or turbine by the equation:

$$\text{D. H. P.} = \frac{\text{Kw.} \times 1.34}{E}$$

in which

E = efficiency of the generator, which may be taken from 0.90 to 0.95, according to type and size.

Example.—A compound condensing engine of 1,000 I. H. P. has a total steam consumption of 16,000 pounds per hour. How does it compare in economy with a turbine using a total of 28,000 pounds of steam per hour attached to a 1,000 Kw. generator having an efficiency of 0.95?

Assuming an efficiency of 0.9 for the engine, the D. H. P. is found to be $1,000 \times 0.9 = 900$; and the water-rate $16,000 \div 900 = 17.8$ pounds per D. H. P. Reducing the electrical output from the generating unit to D. H. P. furnished by the turbine, we have

$$\frac{1,000 \times 1.34}{0.95} = 1,411$$

which calls for $28,000 \div 1,411 = 19.8$ pounds of steam per D.H.P. per hour, from which a comparison of the steam economy of the two machines may be made.

The water-rate of an engine depends to a considerable extent upon the conditions under which it operates, and for accurate results figures should be obtained from tests made upon the same type and size of machine working under practically the same conditions. In the absence of this data builders will usually guarantee a certain steam economy for the conditions given.

Tables LXIV. and LXV. have been made up from figures already given in the text, and while these water-rates are somewhat higher than are often secured with the best makes of engines operating under the most favorable conditions, they represent more nearly the results obtained in actual practice with engines of medium size. The steam consumption of a non-condensing compound engine per unit of power will range from 80 to 85 per cent of that of a simple engine of similar type and size, and from 70 to 75 per cent in the case of a condensing engine.

The economy of compound engines, as given in the following tables, is based upon these assumptions.

TABLE LXIV.

WATER-RATE OF NON-CONDENSING ENGINES.

Type of engine.	Pounds of steam per I. H. P. per hour at full load.	
	Simple.	Compound.
High-speed, single-valve.	30	25
High-speed, four-valve.	28	23
Medium-speed, four-valve.	26	21
Slow-speed, four-valve.	24	19

TABLE LXV.

WATER-RATE OF CONDENSING ENGINES.

Type of engine.	Pounds of steam per I. H. P. per hour at full load.	
	Simple.	Compound.
High-speed, single-valve.	26	20
High-speed, four-valve.	24	18
Medium-speed, four-valve.	22	16
Slow-speed, four-valve.	20	14

It should be remembered, however, that the quantities given in the tables are average values applying to classes of engines rather

than to individual machines. This is mentioned particularly because the water-rates assumed in the problems which occur in the following chapters, will be found to vary slightly in some cases from those given in the tables, although closely approximating them.

264. *Effect of Load Variation on Water-Rate.*—Engines of various types are designed to operate most economically at a given ratio of expansion. When working at constant speed and initial pressure, any reduction of load in case of an automatic engine is offset by shortening the cut-off, thus increasing the expansion and temperature range, which results in lowering the steam economy through increased cylinder condensation and decreased mechanical efficiency. The simple high-speed engine does not fall off very much in economy down to one-half load, but beyond that it drops rapidly. This effect is more marked in the case of a compound engine as will be noticed in Table LXVI. With condensing engines the effect of underloading is not quite so great, but for all approximate work the following table may be used for both non-condensing and condensing engines. The effect of overloading is similar to that of underloading. In some cases the curve of economy is practically symmetrical for a considerable distance each side of the point of full load, while in others it varies somewhat.

TABLE LXVI.

EFFECT OF LOAD VARIATION ON STEAM ECONOMY OF NON-CONDENSING ENGINES.

Type of engine.	Ratio of steam consumption per I.H.P. to that at full load for different under loads.		
	$\frac{3}{4}$ load.	$\frac{1}{2}$ load.	$\frac{1}{4}$ load.
Simple, high-speed.....	1.01	1.07	1.23
Compound, high-speed.....	1.03	1.21	1.78
Compound, slow-speed.....	1.09	1.32	1.88

Example.—If the water-rate of a simple high-speed engine is 30 pounds per I. H. P. per hour, at full load, what will it be at three-fourths, one-half and one-fourth load?

Solution:

$$30 \times 1.01 = 30.3 \text{ pounds at } \frac{3}{4} \text{ load.}$$

$$30 \times 1.07 = 32.1 \text{ pounds at } \frac{1}{2} \text{ load.}$$

$$30 \times 1.23 = 36.9 \text{ pounds at } \frac{1}{4} \text{ load.}$$

The effect of load variation should always be taken into account when proportioning the size of units for a given plant and also when computing the cost of operation or available exhaust in problems involving heating and ventilation.

265. Heat Consumption.—While the steam consumption is the most common measure of the economy of an engine or turbine it does not give a uniform basis for purposes of comparison owing to variations in the initial and back pressures and the percentage of moisture present. When an accurate result is required, the performance should be reduced to heat units per unit output, measured above the maximum temperature of the exhaust. The method of making a comparison of this kind is best shown by a practical illustration.

Example.—A simple high-speed engine supplied with steam at 85 pounds gauge pressure, and 98 per cent quality, exhausts against a back pressure of 1 pound gauge. The water-rate under these conditions is 34 pounds per D. H. P. per hour.

A compound condensing engine supplied with steam at 160 pounds gauge pressure and 200 degrees superheat, requires 18 pounds of steam per D. H. P. per hour. Condenser pressure 2 pounds absolute. Compare the economy of the two engines upon the basis of heat consumption as well as steam consumption.

Taking the first case, we find from Table LXII. that

$$H = (888 \times 0.98) + 298 = 1,168 \text{ (see notation in Paragraph 261.)}$$

$$q = 184.$$

$$\text{Heat supplied per hour per D. H. P.} = 34(1,168 - 184) = 33,456 \text{ B. T. U.}$$

In the second case, we have (Tables LXII. and LXIII.),

$$H = 1,196 + (200 \times 0.541) = 1,304.$$

$$q = 94$$

$$\text{Heat supplied per hour per D. H. P.} = 18(1,304 - 94) = 21,780 \text{ B. T. U.}$$

Comparing the economy of the engines by the two methods, we have

$$\text{Ratio of steam consumption per D. H. P.} = \frac{34}{18} = 1.89$$

$$\text{Ratio of heat consumption per D. H. P.} = \frac{33,456}{21,780} = 1.54$$

When based on the steam consumption the non-condensing engine requires 1.89 times the fuel per D. H. P. as the condensing engine, but on a heat basis, only 1.54 times. If the exhaust is thrown away in each case the comparison should be made on a steam basis, but if the heat in the exhaust can be utilized the latter method gives the true measure of economy.

266. Available Heat in Exhaust Steam.—The proportion of heat discharged in the exhaust varies with the type of engine and the conditions under which it operates. With the engines used in isolated plants it commonly runs from about 80 per cent of that supplied in the best types, up to 90 per cent in the poorer ones. In the absence of definite data it is customary to assume that from 80 to 85 per cent of the steam supplied to the engine is available for heating purposes at exhaust pressure. In the following problems the higher figure has been used.

The remaining portion is condensed in the cylinder and passes out as spray.

267. Effect of Condensing.—While the use of a condenser is limited in combined power and heating plants to summer use and special cases, it is well to consider the economical value of condensing, especially in connection with manufacturing plants, where the cooling water can be secured from a nearby lake or river.

As the effect of a condenser is to lower the back pressure on the engine it may evidently be used either to increase the power with a given steam consumption or to reduce the water-rate per unit of power delivered. When used for summer service, where the power requirements are practically the same throughout the year, steam economy is the object usually sought.

The pressure within the condenser, or degree of vacuum, is commonly expressed in inches of mercury, referred to a barometric pressure of 30 inches.

For example, a 26-inch vacuum means an absolute pressure within the condenser of $30 - 26 = 4$ inches of mercury or $4 \times 0.5 = 2$ pounds per square inch. In like manner a 28-inch vacuum means a pressure of $(30 - 28) \times 0.5 = 1$ pound per square inch. The factor of conversion for inches of mercury to pounds per square inch is actually 0.49 for mercury at a temperature of 60 degrees, but 0.5 is sufficiently accurate for all ordinary work.

The vacuum commonly maintained in average practice, with reciprocating engines, will run from 22 to 26 inches, depending upon various operating conditions. The lower figure corresponds to that obtained by the equipment usually employed with engines, under 100 H. P., and the higher for larger sizes.

Turbines are commonly operated at 27 to 28 inches of vacuum in commercial plants while tests are often run with 29 inches.

The gain in economy by the use of a condenser will depend upon the degree of vacuum maintained and the mean effective pressure of the engine when running non-condensing. This, under ordinary conditions, should theoretically, vary from 20 to 30 per cent, but is found to be considerably less in actual practice.

Table LXVII. is based upon a large number of tests on different types of engines and shows about what saving in steam per unit of power developed may be expected when operating under actual conditions.

The water-rates in Table LXV. were computed from Table LXIV. by the use of ratios taken from Table LXVII.

TABLE LXVII.

SAVING BY OPERATING CONDENSING.

Type of engine.	Percentage of saving in steam by running condensing.	
	Simple, per cent.	Compound, per cent.
High-speed.....	12 to 15	15 to 18
Medium-speed.....	15 " 18	18 " 21
Slow-speed.....	18 " 21	21 " 24

268. Effect of Superheated Steam.—When steam economy is an essential feature a considerable saving may be made by the use of superheated steam. It has not been used in this country to any

great extent in reciprocating engines in the past owing to the difficulty experienced in lubricating the cylinder and to the unequal expansion produced at such high temperatures. However, with more recent improvements in design and construction it is now possible to utilize the advantages of superheat to a considerable extent.

Turbines, on the other hand, do not require lubrication in the steam chambers, as there are no rubbing surfaces, and are therefore usually operated with superheated steam.

In American practice the degree of superheat commonly employed runs from 100 to a maximum of about 160 degrees, being limited to a total temperature of approximately 500 degrees for engines of ordinary construction. Beyond this point the expense of maintaining the superheat, increased cost of construction and repairs soon offset any gain in economy.

In Europe a considerably higher degree of superheat is in common use.

The saving in steam consumption depends principally upon the reduction in cylinder condensation and varies with the operating conditions. Tests of a 250 horse power simple non-condensing engine show a saving of 20 per cent with 150 degrees of superheat, and 25 per cent with 225 degrees. These results correspond to a saving of 1 per cent for each 7.5 degrees rise at the lower temperature and for each 9 degrees at the higher.

A compound condensing engine of the same power shows savings of 15 and 20 per cent for the same relative conditions, corresponding to a gain in economy of 1 per cent for each 10 degrees rise at the lower temperature and for each 11.2 degrees at the higher. A comparison of these results indicates a falling off in effectiveness as the temperature increases, and a greater gain in case of the simple engine than with the compound, both of which are borne out in practice.

The above figures simply show the reduction in water-rate without reference to the cost of superheating. The net gain in economy is usually somewhat less than one-half the apparent gain when water-rate only is considered.

269. Initial Pressure.—Theoretically the efficiency of an engine increases with the temperature range. In actual practice this does not hold true because a point is reached where the cylinder

losses more than offset any gain in steam economy. Furthermore, the cost of construction and maintenance increases rapidly at the higher pressures.

Practical experience has fixed certain limits which are usually followed in the design and construction of different types of machines. These are given in Table LXVIII.

TABLE LXVIII.

INITIAL PRESSURE RANGE FOR RECIPROCATING ENGINES.

Type of engine.	Initial pressure range, (pounds gauge).
Simple high-speed.....	80 to 120
Simple low-speed.....	70 " 110
Compound high-speed.....	120 " 160
Compound low-speed.....	150 " 190

270. Ratio of Expansion.—This, like the initial pressure, has been fixed by experience and may be taken as follows for different types of engines:

Simple non-condensing.....3 to 4 expansions.

Simple condensing.....4 to 5 expansions.

Compound non-condensing...10 to 12 expansions.

Compound condensing.....14 to 18 expansions.

The ratio of expansion in a compound engine is given by the formula

$$\frac{R}{c}$$

in which

R = ratio of volume of low-pressure cylinder to volume of high-pressure cylinder.

c = point of cut-off in high-pressure cylinder.

Power and Size of Engines.

271. Computing Horse Power.—The power of an engine may be determined approximately by the formula

$$\text{I. H. P.} = \frac{PLAN}{33,000}$$

in which

I. H. P. = indicated horse power.

P = mean effective pressure = $[(p \times f) - b] \times d$

p = *absolute* initial steam pressure.

f = factor from Table LXIX. for the given cut-off and clearance.

b = *absolute* back pressure = 16 to 17 pounds for non-condensing engines and 2 to 3 pounds for condensing.

d = $\begin{cases} 0.70 \text{ for simple high-speed engines.} \\ 0.85 \text{ for simple medium-speed engines.} \\ 0.90 \text{ for simple slow-speed engines.} \\ 0.65 \text{ for compound high-speed engines.} \\ 0.80 \text{ for compound medium-speed engines.} \\ 0.85 \text{ for compound slow-speed engines.} \end{cases}$

L = length of stroke, in feet.

A = area of piston, in square inches.

N = number of strokes per minute = r. p. m. $\times 2$.

TABLE LXIX.

FACTORS FOR M. E. P. UNDER DIFFERENT CONDITIONS.

Cut-off at		$\frac{1}{10}$ or 0.1	$\frac{1}{9}$ or 0.111	$\frac{1}{8}$ or 0.125	$\frac{1}{7}$ or 0.143	$\frac{1}{6}$ or 0.167	$\frac{1}{5}$ or 0.2	$\frac{1}{4}$ or 0.25	$\frac{1}{3}$ or 0.333	$\frac{2}{8}$ or 0.375	$\frac{2}{5}$ or 0.4	$\frac{1}{2}$ or 0.5
Per cent clearance	0	0.330	0.355	0.385	0.421	0.465	0.522	0.596	0.699	0.743	0.766	0.846
	1	0.344	0.368	0.397	0.432	0.475	0.530	0.603	0.704	0.747	0.770	0.849
	2	0.357	0.379	0.408	0.442	0.484	0.538	0.609	0.708	0.750	0.773	0.851
	3	0.369	0.391	0.419	0.452	0.493	0.545	0.615	0.712	0.753	0.776	0.853
	4	0.381	0.402	0.429	0.461	0.502	0.552	0.621	0.716	0.756	0.779	0.855
	5	0.392	0.413	0.439	0.470	0.510	0.559	0.626	0.720	0.759	0.782	0.857
	6	0.403	0.423	0.448	0.479	0.517	0.566	0.631	0.724	0.762	0.785	0.859
	7	0.413	0.433	0.457	0.487	0.524	0.572	0.636	0.728	0.765	0.787	0.861
	8	0.423	0.442	0.466	0.495	0.531	0.578	0.641	0.731	0.768	0.789	0.862
	9	0.432	0.451	0.474	0.503	0.538	0.584	0.646	0.734	0.770	0.791	0.863
	10	0.441	0.460	0.482	0.510	0.545	0.590	0.651	0.737	0.772	0.793	0.864
	11	0.450	0.469	0.490	0.517	0.552	0.595	0.655	0.740	0.774	0.795	0.865
	12	0.458	0.476	0.497	0.524	0.559	0.600	0.659	0.743	0.776	0.797	0.866
	13	0.466	0.483	0.504	0.531	0.565	0.605	0.663	0.746	0.778	0.799	0.867
	14	0.474	0.490	0.511	0.538	0.570	0.610	0.667	0.749	0.780	0.801	0.868
	15	0.482	0.497	0.518	0.544	0.574	0.615	0.671	0.752	0.781	0.803	0.869

When the clearance is not known, it may be taken as follows: High-speed engines, 15 per cent; medium-speed engines, 8 per cent; slow-speed engines, 5 per cent. These figures are rather higher than will be found in the best makes, but are on the side of safety.

Example.—What will be the approximate horse power of a simple high-speed non-condensing engine under the following conditions:

Diameter of cylinder.....	14 inches
Length of stroke.....	12 inches
Clearance.....	15 per cent
Initial steam pressure.....	100 pounds, gauge
Back pressure.....	1 pound, gauge
Cut-off.....	$\frac{1}{4}$ stroke
R. p. m.....	250

From the conditions of the problem we have

$$P = [(115 \times 0.671) - 16] \times 0.7 = 43.$$

$$L = 1.$$

$$A = 154.$$

$$N = 500.$$

Substituting these in the formula, and solving we have

$$\text{I. H. P.} = \frac{43 \times 1 \times 154 \times 500}{33,000} = 100$$

In the case of a compound engine the total ratio of expansion is determined by the method previously given, and the problem then treated as though it were a simple engine with cylinder dimensions corresponding to those of the low-pressure cylinder and a point of cut-off equal to the reciprocal of the total ratio of expansion.

272. Engine Capacity.—Methods for determining the power of engine for driving electric generators, refrigerating machines, elevators, line shafting, etc., have been considered briefly in Chapter V. and the results given in Table XLIX.

Steam Turbines.

The principle of the steam turbine and details of construction are fully treated in other books and will not be taken up in the present volume. Special attention, however, will be given to its uses and selection, together with a certain amount of general data relating to economy and installation.

273. Classification of Turbines.—Steam turbines are of three general types, known as the impulse, reaction, and combined impulse and reaction. All turbines, however, make use of both the impulse and reaction principles in varying degrees, and the real

distinction is the manner in which the steam is expanded, this being completed in the nozzles in the impulse turbine, while a portion takes place in the passages of the wheel in the reaction type.

Impulse turbines are divided into *simple* and *compound* machines, and these in turn into *multiple-stage* turbines. Sections through different types of machines are shown in Figs. 126 to 131 inclusive. A simple turbine has a single wheel with a single set

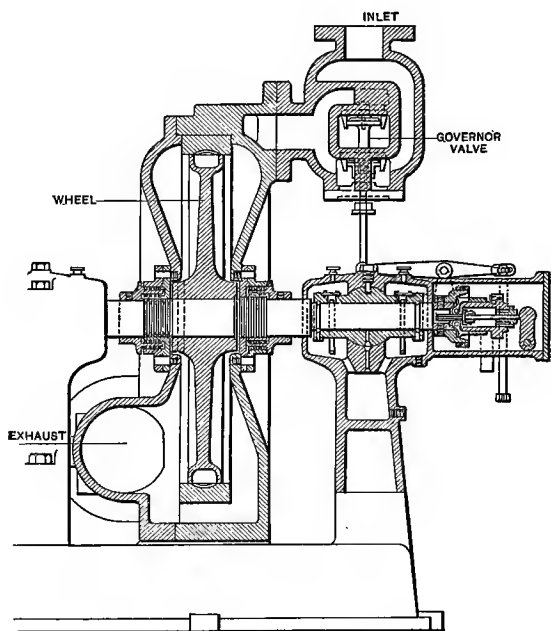


Fig. 126. Simple Impulse Turbine.

of blades, while a compound turbine makes use of two or more wheels upon the same shaft with stationary blades between. This results in lowering the speed a definite amount, according to the number of wheels, which is an important feature in transmitting the power to other machinery.

Multiple-stage turbines are made up of a series of machines attached to a common shaft, each group of wheels and guides being enclosed in a separate chamber and so arranged that the ex-

haust from the first enters the nozzles of the second, and so on to the end of the series.

The various features noted above are combined in practical form according to the service required and include the high-pressure, low-pressure, mixed-pressure and bleeder turbines.

274. *Use of Turbines in Isolated Plants.*—The steam turbine is coming into general use in isolated plants for the driving of electric generators, pumps of various kinds, also blowers, air-compressors, etc. Machines employed for this purpose are of comparatively small size and are usually of the impulse type, although reaction turbines are often used for the generation of

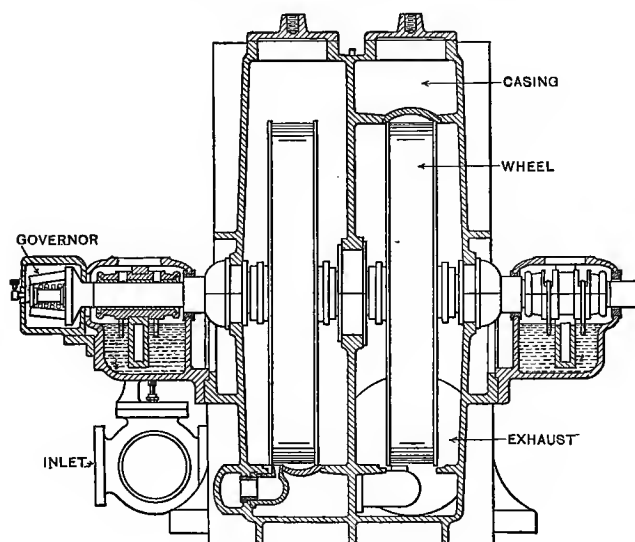


Fig. 127. Impulse Two-Stage Turbine.

power in large manufacturing plants. The chief advantages of the turbine in this class of work are the small floor space required; less weight, doing away with the heavy foundations necessary for reciprocating engines, and the absence of vibration; all of which are important in the case of basement plants in large buildings.

275. *High-Pressure Turbine.*—This is the standard type for power work in the small and medium sizes and may be operated

either non-condensing or condensing as desired. This machine is used under the same general conditions as a reciprocating engine of the same power and a choice between the two is often a matter of local convenience depending upon one or more of the items noted above.

276. Low-Pressure Turbine.—This machine is used chiefly in central station work and large power plants and is designed to take the exhaust steam from a reciprocating engine and expand it down to a low pressure. It is placed between the engine and condenser and adds decidedly to the steam economy as compared with that of the engine and condenser operating alone.

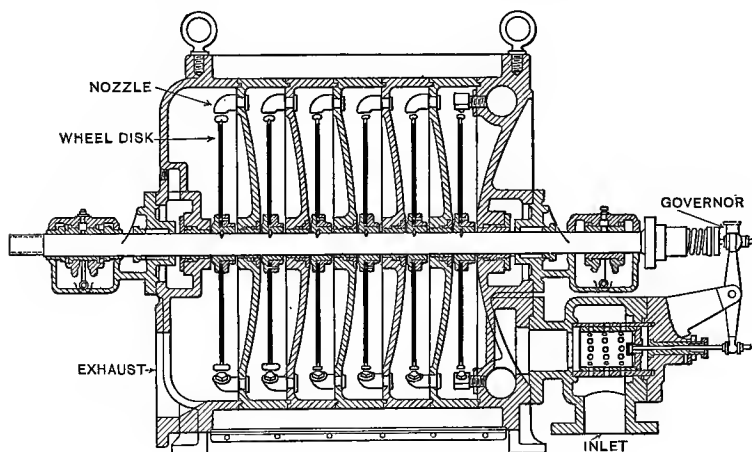


Fig. 128. Compound Impulse Six-Stage Turbine.

277. Mixed-Pressure Turbine.—This is employed under similar conditions to that described above, but in cases where the supply of exhaust is not constant or is insufficient to generate the power required of the turbine. The chief mechanical difference between the mixed-pressure and low-pressure turbine is an arrangement for automatically supplying high-pressure steam to the former when the exhaust falls below the normal requirements. Turbines of this type are usually designed to develop their full power on either high or low-pressure steam or a combination of the two in varying proportions.

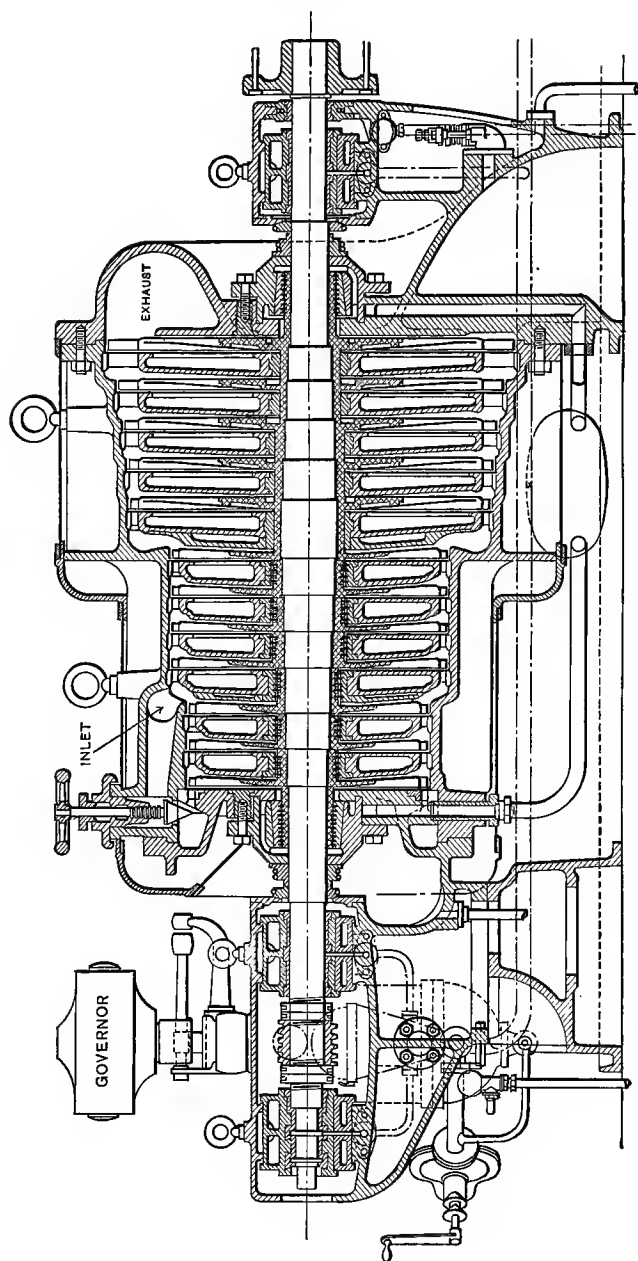


Fig. 129. Compound Impulse Three-Stage Turbine.

278. Bleeder Turbine.—This type of machine is especially adapted to cases where it is desired to use large volumes of steam at a reduced pressure for heating or for manufacturing purposes. While similar in general construction to the high-pressure turbine it is so designed that steam may be drawn from an intermediate stage after having performed work in passing through the high-pressure stages. It may be set to deliver steam into the heating system at any desired pressure within its range and operates automatically according to the volume of steam required.

279. Turbine Compared With Engine.—Some of the advantages of the steam turbine as applied to isolated plants have already been mentioned. Economy of space is always important in the case of office buildings and department stores, and the fact that the turbine is of comparatively light weight and produces no vibration makes it possible to carry it on a steel beam and concrete construction in many cases, thus leaving a free space below for the accommodation of condensers and heavy piping. Among other advantages are simplicity of construction, which calls for minimum attention; absence of oil in the steam spaces, thus adapting it to the use of superheated steam and also making the exhaust suitable for heating purposes and the condensation available for boiler feeding without purification; close speed regulation; and high efficiency for a wide variation in load.

Owing to its comparatively high speed it is not so well adapted to the driving of direct-current generators, air compressors, refrigerating machines, etc., as the reciprocating engine, but for alternating-current generators, centrifugal pumps, blowers, or any other machine requiring a high speed the turbine is to be preferred. In the matter of expense the turbine has the advantage over the reciprocating engine, both in cost of installation and maintenance.

280. Thermal Efficiency.—The thermal efficiency of a turbine is computed in the same manner as for a reciprocating engine, using the expression $\frac{2,546}{W(H-q)}$ already given in Paragraph 261.

281. Steam Economy.—For the sizes used in isolated plants there is very little difference in the steam economy of turbines and reciprocating engines of equal grade. In case of the larger

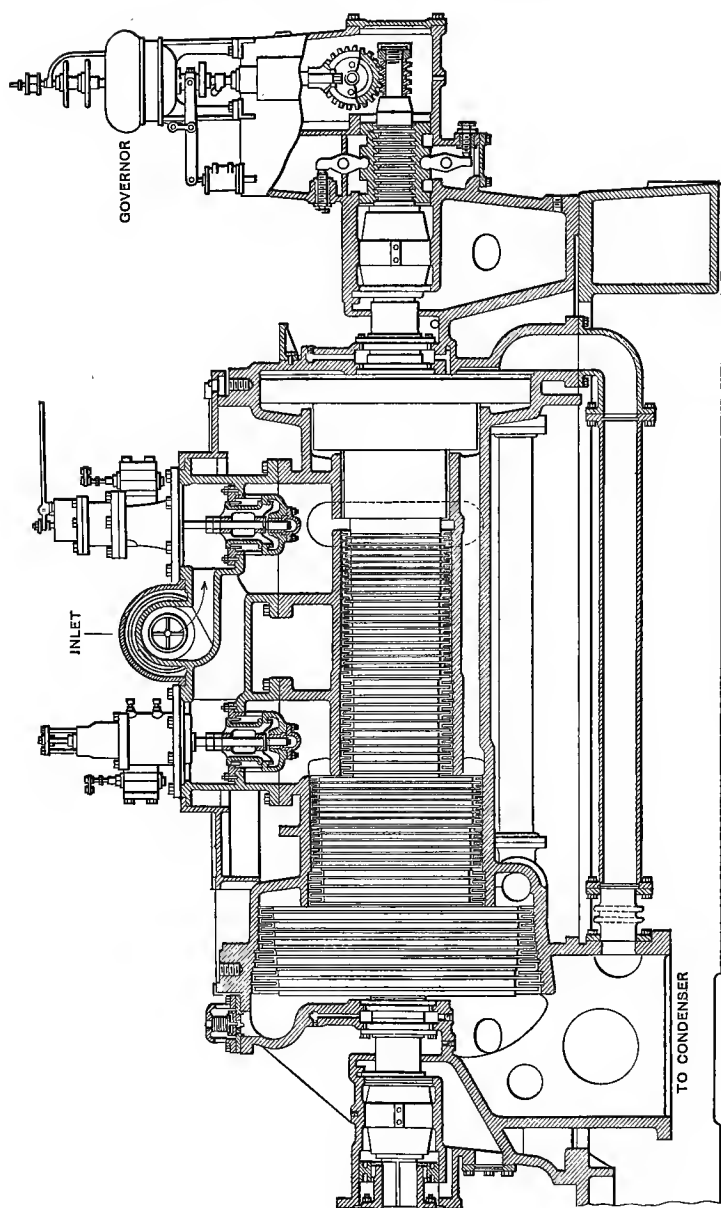


Fig. 130. Reaction Turbine.

units, say above 1,500 or 2,000 kilowatts capacity, the turbine will be found more economical in the use of steam than the engine, but with those of small size the simple high-speed non-condensing engine will be found slightly more efficient than a turbine operating under the same conditions. The steam consumption for a given capacity is so nearly alike for the two machines that the matter of selection is usually based upon other considerations.

282. Effect of Condensing.—The turbine is much more sensitive to the effect of a vacuum than a reciprocating engine owing to the greater number of expansions. This may be illustrated by two engines having cylinders of the same diameter but with the stroke of the second twice that of the first. Let the first cut off at one-fourth stroke, and the same volume and weight of steam

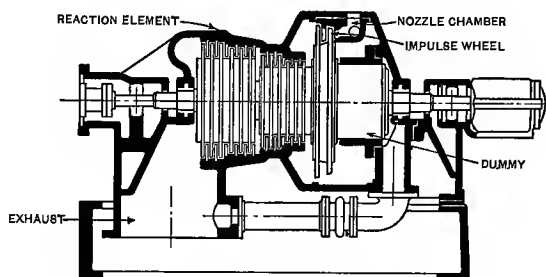


Fig. 131. Combined Impulse and Reaction Turbine.

be used by each engine per stroke. It is evident from the above that the first will have four expansions and the second eight. If now 10 pounds be added to the mean effective pressure in each case by attaching a condenser, the gain in actual power will be twice as great in the second as in the first, because the stroke is twice as long. In other words, with a given set of conditions, the gain by condensing varies theoretically as the number of expansions.

The best types of compound condensing engines average about sixteen expansions while a turbine will easily expand the steam one hundred times or more. Hence, under the above conditions, the relative effect of adding an inch to the vacuum will be from six to eight times as great in the case of a turbine as with an

engine. Furthermore, there is an increase in efficiency due to the greater expansion of the steam.

Very little is to be gained by increasing the vacuum beyond 26 inches in case of a reciprocating engine, while with a turbine a decrease of 3 to 4 per cent in steam consumption may be obtained for each inch increase in vacuum between 25 and 29 inches.

Comparing the Performance of Engines and Turbines.

283. Methods of Comparison.—The performance of reciprocating engines and turbines may be compared in different ways according to the degree of accuracy required. For rough approximations the pounds of steam per brake horse power or kilowatt per hour may be taken without reference to the conditions of temperature and pressure, but for more exact comparisons it is necessary to reduce all working conditions to a common basis. Manufacturers of turbines usually furnish a set of correction curves showing the effect on the steam economy by changes in the initial pressure, back pressure, and temperature of the steam. By means of these it is a simple matter to reduce the actual water-rate of the turbine, as determined by test, to what might be expected under working conditions identical with those of the engine. Another method of comparison is by means of the heat consumption, per unit output, as already described in Paragraph 265.

Size of Generating Units.

284. Conditions for Highest Efficiency.—In order to obtain the highest efficiency of an engine, turbine or generator it is necessary that it shall operate at approximately full load, hence care should be taken in each case to so proportion the units that the above condition may be approached as nearly as possible at all times. In manufacturing plants where the power requirements are fairly uniform this is a comparatively simple matter, but where the load varies from hour to hour, as frequently happens in different types of buildings, considerable care must be exercised in selecting the proper size of units. It is, of course, impossible to secure ideal conditions at all times, and the plant

must not be complicated by the use of too many machines of different size. In designing a plant a middle course should be chosen, giving due regard both to load variation and simplicity of equipment, providing in all important cases one or more duplicate units for use in case of accident or necessary repairs. Practical illustrations along this line are given in the various problems in Chapter XII.

CHAPTER VIII.

AUXILIARY EQUIPMENT.

This includes the necessary equipment for supplying the plant with water for all purposes, also fire pumps, feed pumps and injectors, vacuum pumps, heaters, and condensers.

Water Supply.

285. Different Uses.—The purposes for which water is required in any given case will depend upon the type of building. In general, these may be included under the heads of boiler feeding, condenser service, domestic purposes, manufacturing processes, air conditioning, and fire protection.

The first step in providing a water supply for a power or industrial plant is to determine the maximum quantity required in a given time. This should be made sufficiently large to cover any unexpected demands within reasonable limits, without overtaxing the supply.

286. Boiler Feeding.—Under normal conditions this will amount to approximately 35 pounds per boiler horse power per hour. Increasing this from 30 to 50 per cent for possible overloads, or 40 per cent for average conditions, calls for

$$\frac{35 \times 1.4}{8.3} = 6 \text{ gallons per hour per rated boiler horse power.}$$

In case of factories and shops, where the power requirements are more uniform, 5 gallons per horse power per hour will usually suffice. Each plant should be considered as a separate problem and provision made for all possible conditions of overloading. The above refers to cases where all of the exhaust steam is wasted. When a portion of the exhaust is utilized for heating purposes of any kind and the condensation returned to the boilers, this may be deducted from the cold feed requirements. When the engines are connected with a surface condenser, and the oil removed from the exhaust, or when turbines are employed, only enough "make-up" water is required to prevent the distilled

water from corroding the boiler plates. About 10 per cent of fresh water is usually ample for this purpose. In case the cooling water is sufficiently pure for steaming purposes, the "make-up" may be taken from the discharge pipe of the condenser, thus eliminating the item of feed water entirely from the total quantity to be provided in a condensing plant.

In city buildings condensers are not usually employed, on account of the cost of cooling water when taken from the mains, and the quantity of "make-up" water must be computed in connection with the heating system as previously noted. It should be remembered, however, that the heating plant is shut down in the summer and in many cases the entire exhaust discharged outboard, while in others the engines may be run condensing during the non-heating season.

In all cases the conditions requiring the maximum amount of fresh feed water should be taken into account when determining the capacity of the supply.

287. Condensing.—While the quantity of cooling water required varies with the initial and final temperatures and the vacuum carried, it is customary, in approximating the water supply, to allow 30 pounds per pound of steam condensed for jet and barometric condensers, and 35 pounds for surface condensers, when used in connection with reciprocating engines. For turbines, twice that amount should be supplied.

The weight of steam to be condensed will depend upon the type and size of the engines, and in the absence of more exact information, may be taken from Table LXV.

288. Domestic and Manufacturing Purposes.—No general rule can be given for the quantity of water required for these purposes and it must be determined in each case from whatever data is available and added to the total supply.

289. Air Conditioning.—When a spray washer is used for filtering purposes only, the water required is simply that necessary to replace evaporation losses, with an occasional flushing of the tank. Under these conditions it is not usually necessary to make any additional allowance on this account.

When city or artesian well water is used for air cooling the amount required is greatly increased.

It has been stated in Paragraph 164, that when the tempera-

ture of the spray is about 3 degrees below the dew-point of the entering air no evaporation will take place and the entire cooling effect is produced by the transmission of heat from the air to the water. Assuming a water temperature of 55 degrees and an air temperature of 80 degrees, this condition will prevail for all relative humidities above 50 per cent. For an air-cooling effect of 10 degrees, with a rise in water temperature of 20 degrees, it will require, approximately, 1.1 gallons per 1,000 cubic feet of air. Water for this purpose is usually taken from an artesian well, and if the flow is not sufficient to produce the desired cooling effect it may be supplemented by the addition of city water.

290. Fire Protection.—In the modern fire-proof power house very little is done in the way of fire protection except in the construction of the building itself. With industrial plants, institutions, and city buildings of various kinds, a system of hydrants and sprinklers is an important item in the matter of water supply. The details to be followed in any particular locality are governed by the underwriters' regulations for that district, but there are certain general rules which may be followed in computing the water supply for approximate work. Unless local regulations require different treatment, 250 gallons per minute should be provided for each $1\frac{1}{8}$ -inch fire stream and 125 gallons for each $\frac{3}{4}$ -inch fire stream. Each sprinkler head will require from 6 to 8 gallons per minute according to the pressure, but it will not be necessary to count upon all being in operation at the same time. A record of over 80 fires showed that the number of sprinklers in action at one time did not exceed 15 in any case. In general, a supply of 500 to 1,000 gallons per minute for sprinkler service will prove ample in plants of medium size. In some cases the water supply may be based upon the capacity of the fire pumps. The arrangement of the system also has an important bearing upon the amount of water to be furnished continuously, as in many cases the sprinklers are supplied from a storage tank and only the hydrant service need be considered in this connection. When the requirements for fire protection exceed those for other purposes by a certain amount, they may be taken as the governing factor in fixing the water supply for the entire plant. As but a portion of the machinery would be in operation during a fire, the principal use of water, aside from fire purposes, would

be for the steam used by the pumps, which may be taken as 1 gallon for each 80 to 100 gallons delivered by the fire streams.

291. *Sources of Supply.*—The source of supply in city buildings is necessarily from the service mains unless the location hap-

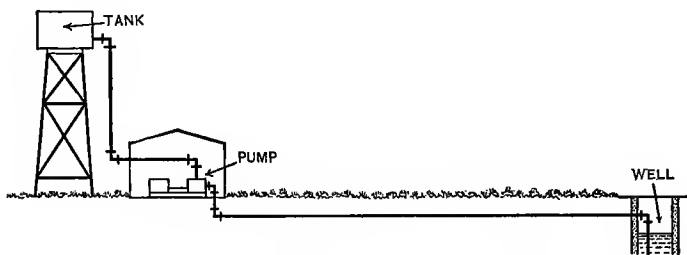


Fig. 132. Location of Pump for Low Elevation.

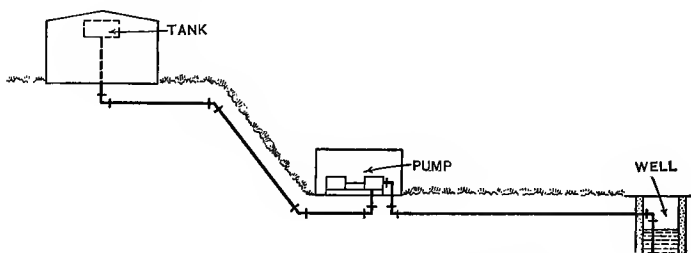


Fig. 133. Location of Pump for High Elevation.

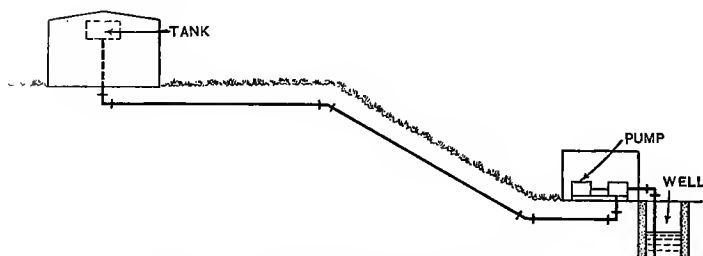


Fig. 134. Pump Located at Source of Supply.

pens to be near a water-front, and even then, the latter supply may only be fit for condensing and fire purposes.

Water for air cooling and similar uses may be obtained from artesian or deep wells located upon the premises, or at least in

sufficient quantity to supplement that drawn from the service mains, as previously noted. In manufacturing plants and institutions located outside of thickly settled sections the water supply may come from lakes, rivers, private reservoirs fed by springs, artesian wells, or public water mains, according to local conditions.

Pumps for General Service.

292. Location of Pump.—When the water is taken from public mains, or flows to the plant by gravity, it is simply a matter

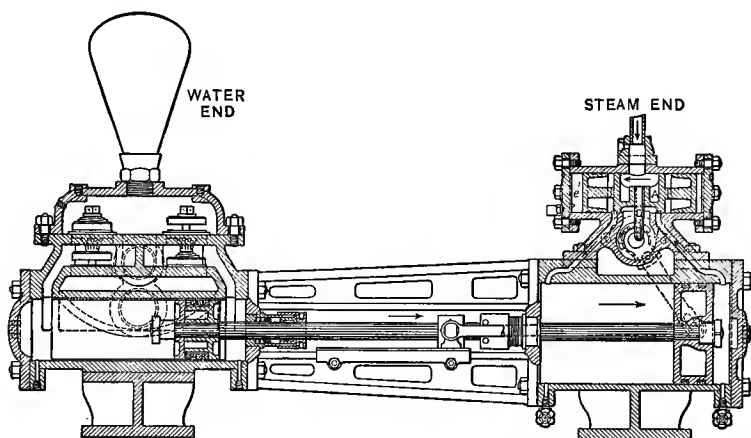


Fig. 135. Section through a Direct-Acting Steam Pump.

of proportioning the size of pipe to the pressure and the volume required.

When the power house is at a higher elevation than the source of supply the water must be pumped. If the conditions are such that the friction head in the pipe, plus the elevation, does not exceed 15 to 18 feet the pumping equipment may be placed in or near the power house as shown in Fig. 132. Otherwise it must be located at some intermediate point where this limit will not be exceeded. Two arrangements of this kind are shown in Figs 133 and 134. In the first of these the grades are such that the pump may be located near the power house, but at a lower level, while in the second it is necessary to place the pump close to the

well or other source of supply. In both cases the water is forced into the tank instead of being drawn up by suction.

293. *Types of Pumps Used.*—Direct-acting steam pumps, engine and turbine-driven plunger and centrifugal pumps are adapted to the first of these conditions, and also to the last when the distance is not so great as to make the carrying of steam from the power house both expensive and wasteful. When the distance exceeds a certain limit it is usually better to drive the pump by an electric motor or gasoline engine than to install and care for a special boiler.

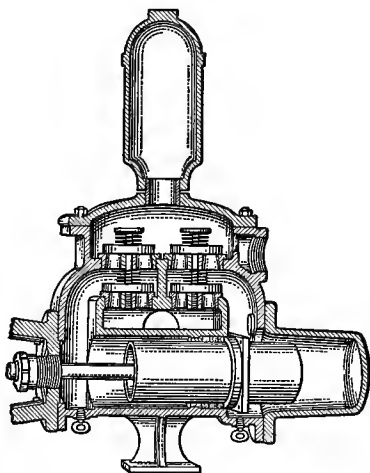


Fig. 136. Section through the Water End of a Plunger Pump.

When the water is taken from a river, and the grades are suitable, a hydraulic ram may be employed in cases where there is an abundance of water. This type of pump is made in a large number of sizes and requires practically no attention, as the only parts subject to wear are the rubber valve disks.

294. *Pumps for Artesian Wells.*—In the case of artesian wells there are two methods in common use for pumping the water to the power house. In the first, each well is equipped with a lift-pump driven by steam, electricity or gasoline, and connected with a common main leading to the power house. The second method

makes use of the "air lift," which is especially adapted to cases where it is desired to increase the flow of water and to plants using a series of wells. With this arrangement one compressing outfit, located in the power house, may be employed for the entire work.

295. Direct-Acting Steam Pump.—This type of pump is made in a variety of forms and sizes and is widely used in connection with power plant work. A typical single piston pump is shown in Fig. 135. Pumps of this kind are adapted to locations where the water is free from grit or other substances likely to destroy the packing. When the above conditions are present, the plunger pump is preferable on account of the ease with which the worn parts may be packed or renewed. A section through the water end of a pump of this type is shown in Fig. 136. One of the chief disadvantages of the direct-acting pump is the excessive steam consumption as compared with an engine or turbine, but this is

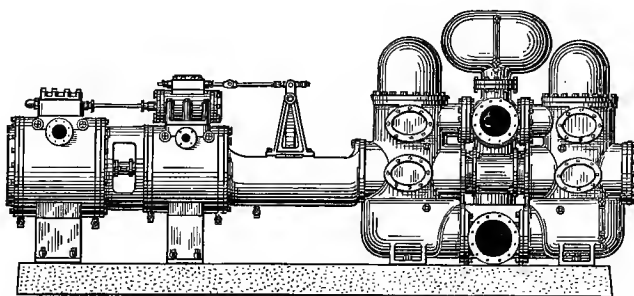


Fig. 137. Compound Plunger Pump.

offset in many cases by the low cost of installation, convenience, and ease with which the speed may be regulated to meet varying requirements. Pumps of this type are made single, duplex, simple and compound, according to the work to be performed. A compound pump, especially adapted to elevator service, is shown in Fig. 137. Attention is called in this cut, as well as in Fig. 151, to the special features of design which allow the plungers to be inspected and repacked from the outside without removing the cylinder heads.

Direct-acting pumps have an average mechanical efficiency of

60 to 80 per cent and a "slippage" of 15 to 20 per cent under ordinary conditions of adjustment. The steam consumption of small and medium-size duplex pumps will run from 80 to 160 pounds per delivered horse power per hour according to size. By compounding, as in Fig. 137, this may be reduced from 40 to 50 per cent. Pumps of this type are operated at a comparatively low speed, although the steam consumption per unit of work de-

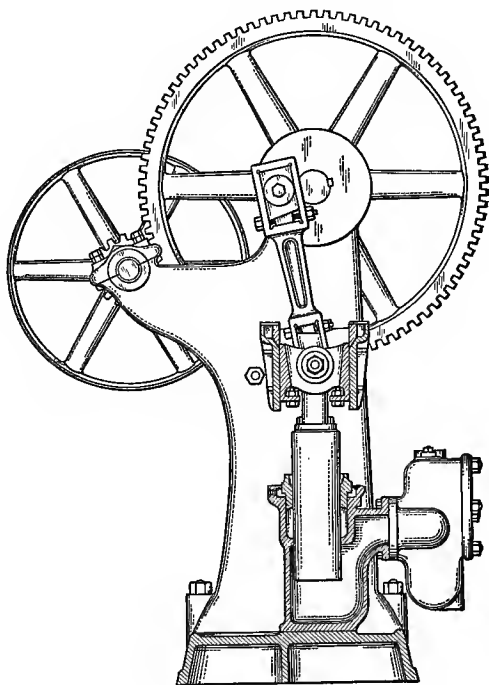


Fig. 138. Geared Power Pump.

creases as the speed increases. For pumps of larger size the piston speed is usually limited to 100 feet per minute, but for strokes of less than 12 or 14 inches the piston speed should be reduced proportionately. Pumps which are to be run continuously should be designed to operate at about one-half the maximum allowable speed noted above.

296. *Power Pumps.*—Power or geared pumps, the general prin-

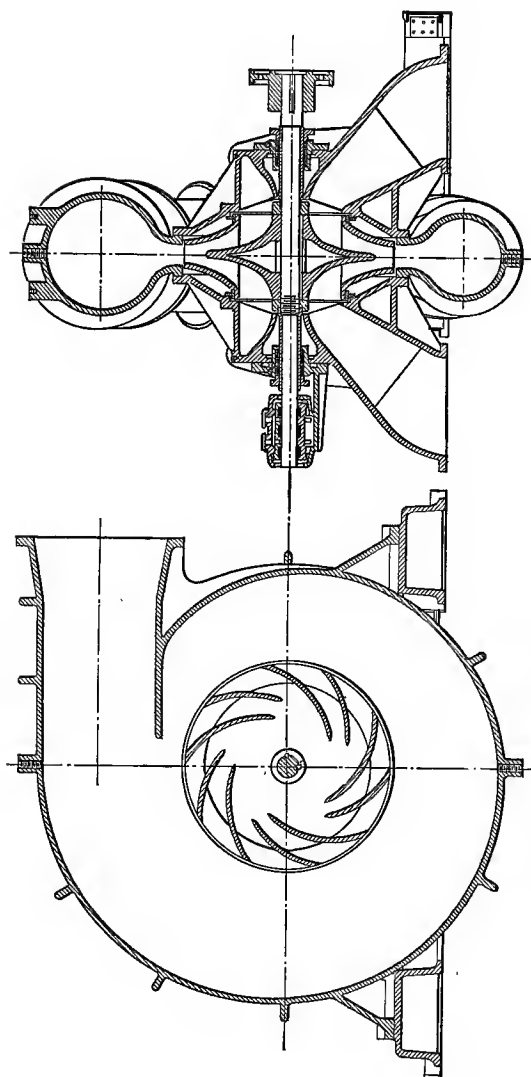


Fig. 139. Sections of a Centrifugal Pump of the Volute Type.

ciple of which is shown in Fig. 138, are used for practically the same purposes as the direct-acting steam pump. They are more economical to operate as they may be driven by an engine, turbine or electric motor as most convenient. When belted to line shafting, or driven by prime movers requiring a constant speed, they are not so desirable as the steam pump owing to the difficulty of regulation. When used for supplying tanks and reservoirs or other purposes where they may be run at constant speed for periods of considerable length they give very satisfactory results and are supplanting the steam pump in many lines of service.

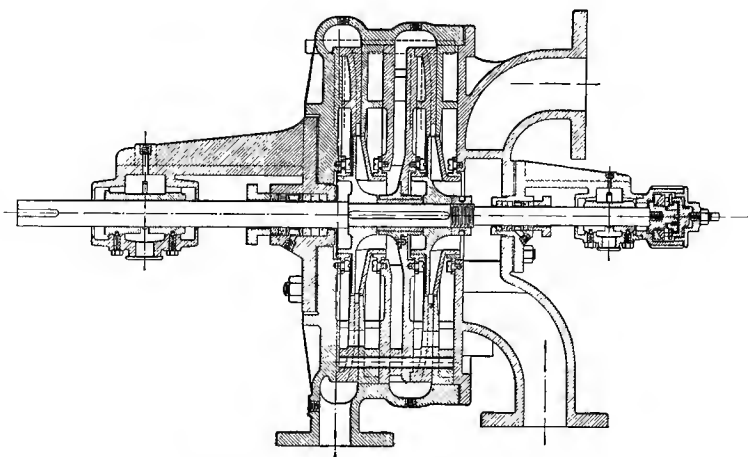


Fig. 140. Section through a Two-Stage Turbine Pump.

The efficiency of the triplex pump may be taken as about 60 per cent for total heads of 100 feet, 70 per cent for 200 feet, and 80 per cent for 300 feet. "Slippage" 15 to 20 per cent.

297. *Centrifugal Pumps.*—Pumps of this type have come into general use for nearly all purposes with the advent of the electric motor and steam turbine. Centrifugal pumps are of two general forms, the *volute* and *turbine*, varying chiefly with the interior construction of the casing.

The volute pump (Fig. 139), is usually of the single-stage design and limited to heads of about 100 to 120 feet, although two-stage machines are constructed for much higher pressures. Tur-

bine pumps are designed for high lifts and are usually compounded in order to limit the peripheral velocity and reduce the friction. A section through a two-stage turbine pump is shown in Fig. 140, and an exterior view of a three-stage direct-connected outfit of this type in Fig. 152. It is important when using a centrifugal pump of any type to select one designed for the conditions under which it is to operate.

The efficiency of centrifugal pumps commonly runs from 60 to 80 per cent for the best types working under the conditions for which they are designed.

Some of the advantages of this pump are simplicity and compactness, absence of valves, low cost, uniform delivery, and high rotative speed, adapting it to direct connection with motors and turbines.

On the other hand it is not possible to obtain so high an efficiency as with the best grades of piston pumps when the latter are kept in first-class condition. Furthermore, the speed cannot be varied, except within narrow limits without loss of efficiency.

298. Deep Well Pumps.—Deep wells are of two general kinds; open wells having a considerable diameter, and driven or artesian wells. The type of equipment required in the first case is shown in Fig. 141 and consists of one or more pump cylinders placed within 18 or 20 feet of the surface of the water and connected with some form of pumping head at the top of the well by means of a long rod. The water is raised to the cylinder by suction and lifted or forced from this point to the surface of the ground.

The pump shown in Fig. 141 is operated by a direct-connected

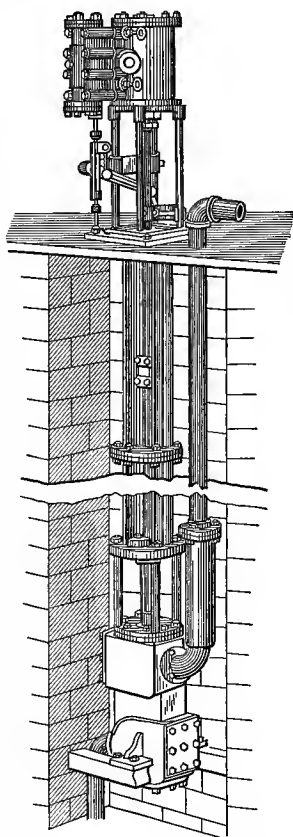


Fig. 141. Deep-Well Pump.

steam cylinder. A pumping head for an electric drive is illustrated in Fig. 142. In this arrangement a second or differential plunger is employed which tends to equalize the flow of water from the discharge pipe. This is shown in the cut just back of the air chamber.

In case of an artesian well, an outer tube is driven to the required depth, extending to the surface of the ground. Inside of this, submerged in the water near the bottom, is the "barrel" containing the pump bucket and foot valve, the general form of which is shown in Fig. 143. The bucket or plunger is connected with a

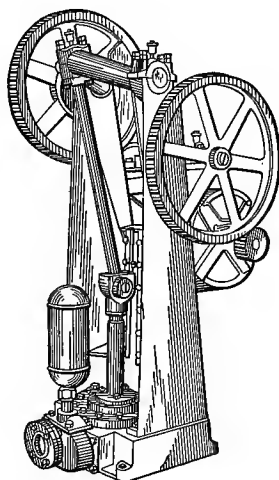


Fig. 142. Pump Head for Electric Drive.

pump head at the top of the well by means of a wooden sucker rod, this material being used in order to reduce the weight. Pumps of this kind may be operated by a direct-acting steam cylinder (Fig. 144), or by a geared electric motor or gasoline engine.

Deep well pumps have an efficiency of 40 to 50 per cent and a "slippage" of 10 to 15 per cent.

299. *Air Lift*.—The principle of the air lift is shown in Fig. 145. A water pipe is carried down to the required depth, together

with an air pipe, either on the outside or inside as most convenient. In operation compressed air is forced into the water pipe near the bottom, thus decreasing the density of the water within it, due to the air bubbles. The flow upward is produced by the difference in weight between the column of solid water *B* and

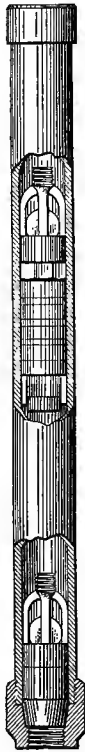


Fig. 143. Pump
Barrel for a
Driven Well.

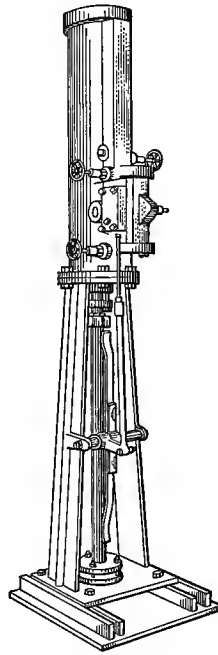


Fig. 144.
Steam-Driven
Pump Head.

the mixture of air and water *C*. Air under sufficient pressure for raising the water is furnished by a compressor located in the power house, the air pipe following the general line of the water pipe.

The tank, *D*, in Fig. 145, is for equalizing the pressure and reducing the pulsation between strokes. Referring again to the cut, *B* is called the *submergence*, *C* the *lift*, and *A* the *total head*.

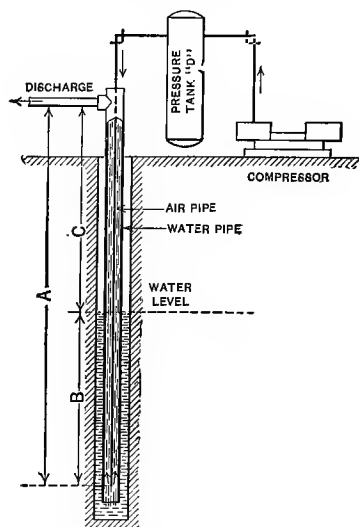


Fig. 145. Diagram of an Air Lift.

In practical work the submergence is expressed as a percentage of the total head. For example, if *A* and *B* are 250 and 150 feet respectively, the submergence is $\frac{150}{250} = 0.60$ or 60 per cent. The efficiency of an air lift increases with the percentage of submergence and commonly runs from about 30 per cent for $\frac{B}{A} = 0.5$ up to 50 per cent for $\frac{B}{A} = 0.8$. A ratio of about 1 to 6 between the areas of the air and water pipes gives the best results for average conditions. If the air pipe is made too large, power will be wasted in producing a high water velocity, and if too small, the air bubbles will not expand sufficiently to fill the discharge pipe and will rise through the water without lifting it.

300. Hydraulic Ram.—This offers the cheapest method of pumping where there is a sufficient supply of water and suitable

grades. A typical ram is shown in section, in Fig. 146, in which *A* is the drive pipe connection, *C* the waste valve, *D* the check valve, *E* the air chamber, and *F* the delivery pipe. In action, water enters *A* and passes out through *C* until sufficient force has been produced in the drive pipe, due to the inertia of the rapidly moving column of water, to close *C* and open *D*. This allows the water to pass into the lower part of chamber *E*, compressing the air above it until sufficient pressure is obtained to close the check *D* and force a certain amount of water through *F* to the desired point. This constitutes a single stroke of the ram which is repeated at regular intervals.

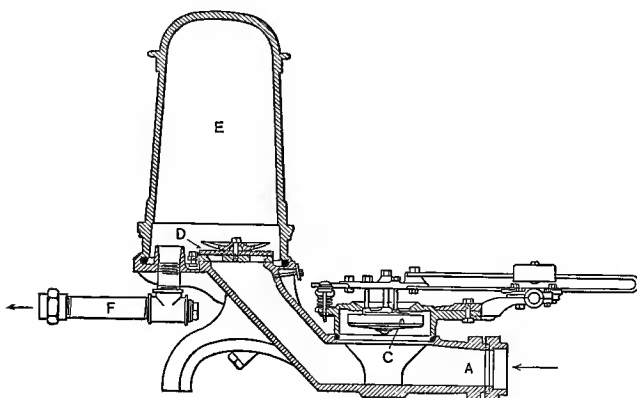


Fig. 146. Section through a Hydraulic Ram.

Two general methods of connecting a hydraulic ram are shown in Figs. 147 and 148. If the drive pipe is too long the excessive friction will interfere with the proper action of the ram and if too short, water will be forced back into the drive tank. In practice it is customary to make the length of drive pipe equal approximately to the height of lift (h) to the tank or reservoir. When it is necessary, for any reason, to locate the ram at a greater distance from the source of supply, the required length of drive pipe can be secured by introducing a stand-pipe or intermediate drive tank nearer the ram as shown in Fig. 148. For large quantities of water the fall from the source to the ram should not be

less than 2 feet, and unless special provisions are made should not in general exceed 12 or 15 feet owing to the shock when the flow is suddenly checked in the drive pipe.

Standard rams are made in large sizes, using from 400 to 15,000 gallons of water per minute (G), operating under a fall of $1\frac{1}{2}$ to

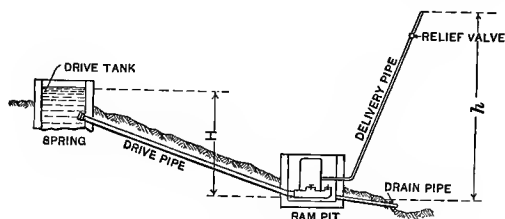


Fig. 147. Connections for a Hydraulic Ram.

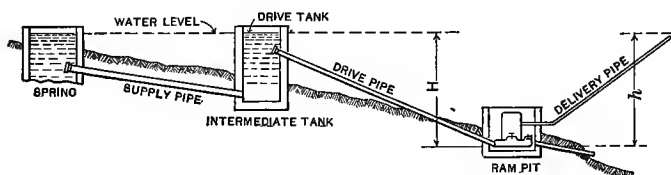


Fig. 148. Hydraulic Ram with Drive Tank.

50 feet, and raising water 35 feet per foot fall, up to a maximum of about 800 feet.

The working formulae for the hydraulic ram are as follows:

$$g = \frac{2 \times G \times H}{3 \times h}$$

$$G = \frac{3 \times h \times g}{2 \times H}$$

$$H = \frac{3 \times h \times G}{2 \times G}$$

$$h = \frac{2 \times G \times H}{3 \times g}$$

in which

g = gallons discharged by the ram.

G = gallons required for operating the ram, (including g .)

H = fall, in feet, from source of supply to ram, (see Figs 147 and 148).

h = height, in feet, to which water is lifted above the ram (see Figs. 147 and 148).

Capacity of Pumps.

301. *Methods of Computing.*—The capacity of a piston or plunger pump is given by the formula

$$G = D \times L \times N \times S \times C$$

in which

G = gallons discharged per minute.

D = piston displacement per inch of stroke (Table LXX.).

L = length of stroke, in inches.

N = number of strokes per minute.

S = factor for "slippage," (Table LXXI.).

C = number of cylinders.

Table LXX. has been computed for use with the above formula and gives the displacement in gallons per inch stroke for pistons or plungers of different diameters. Table LXXI. gives "slippage factors" for different types of pumps. These are equal to 1 minus the per cent of slippage, as previously given.

TABLE LXX.

PISTON DISPLACEMENT PER INCH STROKE.

Diameter of piston or plunger, in inches.	Displacement, in gallons per inch stroke.	Diameter of piston or plunger, in inches.	Displacement in gallons per inch stroke.
4	0.055	10	0.34
5	0.085	12	0.49
6	0.122	14	0.66
7	0.166	16	0.87
8	0.218	18	1.10
9	0.285	20	1.36

TABLE LXXI.

EFFECT OF "SLIPPAGE" ON PUMP CAPACITY.

Type of pump.	"Slippage" factor.
Deep well.....	0.88
Medium piston or plunger.....	0.85
Small piston or plunger.....	0.80
Geared or power.....	0.82

Example.—What will be the capacity of a duplex piston pump, having 12-inch cylinders and a 20-inch stroke, while running at a speed of 50 strokes per minute?

$$0.49 \times 20 \times 50 \times 0.85 \times 2 = 833 \text{ gallons per minute.}$$

In practice it is usually best to obtain tables of pump ratings from the manufacturers, together with guaranteed "slippage factors" for average conditions. This simplifies the process of computation and also places the responsibility of the result upon the makers.

Power Required for Pumping Water.

302. Computations Involved.—The theoretical horse power required for elevating water is given by the formula

$$\text{W. H. P.} = \frac{W \times H}{33,000}$$

in which

W. H. P. = the "water horse power."

W = pounds of water raised per minute.

H = height of lift, in feet.

In practical work the quantity of water handled is usually expressed in gallons. Furthermore, the frictional resistance of the supply and discharge pipes must be considered and also the mechanical efficiency of the pump. Taking these into account we have for a working formula

$$\text{D. H. P.} = \frac{G \times H}{4,000 \times E}$$

in which

D. H. P. = delivered horse power required to operate the pump.

G = gallons discharged per minute.

H = total head = friction head in pipe line plus the lift, both taken in feet.

E = efficiency of pump.

(Pounds per square inch may be reduced to feet head by multiplying by 2.4.)

Example.—What D. H. P. will be required by a pump, having an efficiency of 60 per cent, to raise 160 gallons of water per minute a height of 50 feet through a 4-inch pipe 1,500 feet in length?

The friction head from Table LXXII. is 29.1; making the total head pumped against $(29.1 \times 1.5) + 50 = 94$ feet. Substituting in the formula we have

$$\text{D. H. P.} = \frac{160 \times 94}{4,000 \times 0.6} = 6.3.$$

TABLE LXXII.

FRICTION HEAD IN FEET FOR PIPES 1,000 FEET IN LENGTH.

Size pipe, inches	Rate of flow and friction head.										
1	Gals. per min.	3	4	5	6	8	10	12	14	16	18
	Friction head	12.6	21.4	32.5	45.5	78	117	164	220	280	350
1½	Gals. per min.	5	10	15	20	25	30	35	40	45	50
	Friction head	3.9	14.3	30	52	78	110	147	188	232	284
2	Gals. per min.	25	30	35	40	45	50	60	70	80	90
	Friction head	27.3	3.84	51	66	82	99	139	184	237	294
3	Gals. per min.	50	60	70	80	90	100	120	140	160	180
	Friction head	13.8	19.2	25.7	32.8	40.8	49.6	70	92	118	148
4	Gals. per min.	100	120	140	160	180	200	240	280	320	360
	Friction head	12.2	17.1	22.8	29.1	36.1	44	62	82	105	131
5	Gals. per min.	160	180	200	240	280	320	360	400	450	500
	Friction head	9.8	12.2	14.8	20.8	27.7	35.4	45	54	67	81
6	Gals. per min.	200	230	260	300	340	380	440	500	600	700
	Friction head	6.1	7.9	10	13.5	16.0	20.0	26.1	33.3	46.4	61
8	Gals. per min.	350	400	450	500	600	700	800	900	1,000	1,200
	Friction head	4.2	5.5	6.8	8.2	11.5	15.1	19.5	24.5	29.9	40.0
10	Gals. per min.	700	750	800	900	1,000	1,200	1,400	1,600	1,800	2,000
	Friction head	5.1	5.9	6.6	8.3	10.0	13.8	18.4	23.6	29.9	36.0
12	Gals. per min.	1,400	1,500	1,600	1,700	1,800	2,000	2,200	2,400	2,600	2,800
	Friction head	7.6	8.8	9.7	10.9	12.2	15.0	18.1	21.0	24.1	27.0
16	Gals. per min.	3,000	3,200	3,400	3,600	3,800	4,000	4,500	5,000	5,500	6,000
	Friction head	7.8	8.8	9.9	11.0	12.1	13.2	16.6	20.1	24.0	28.0
20	Gals. per min.	6,000	6,250	6,500	7,000	7,500	8,000	8,500	9,000	9,500	10,000
	Friction head	9.5	10.2	11.2	12.5	14.3	16.1	17.9	20.0	22.2	24.3

303. *Pump Efficiency.*—The mechanical efficiency of a direct-acting steam pump is taken as the ratio of the water horse power to the indicated horse power of the steam cylinder. In the case of power pumps of different kinds, driven by separate engines or motors, the mechanical efficiency is the ratio of the water horse power to the input or delivered horse power. In the air lift the ratio of the water horse power to that required by the air compressor is taken. Table LXXIII. gives in tabulated form the average efficiencies of different types of pumps as already noted in the text.

TABLE LXXIII.

PUMP EFFICIENCIES.

Type of pump.	Average mechanical efficiency, per cent.
Direct-acting steam.	70
Triplex power, 100 feet head.	60
Triplex power, 200 feet head.	70
Triplex power, 300 feet head.	80
Centrifugal.	70
Deep well.	45
Air lift.	40

The steam consumption of a direct-acting pump of the type

commonly used for boiler feeding and elevator service will vary somewhat according to size, but may be taken approximately as given in Table LXXIV.

TABLE LXXIV.
WATER-RATES OF DIRECT-ACTING STEAM PUMPS.

Type of pump.	Pounds of steam per D.H.P. per hour.
Simple non-condensing.....	120
Compound non-condensing.....	65
Triple non-condensing.....	40
High duty non-condensing.....	30

304. *Power for Air Lifts.*—The air pressure required for raising water with an air lift may be found with sufficient accuracy by dividing the actual submergence, in feet, ("B," Fig. 145), by 2.

The volume of air required, at the given pressure, will be the same as the volume of water which it displaces, or in other words, each cubic foot of air delivered at the bottom of the well will raise 7.2 gallons of water.

From the above, and what has already been said regarding air compression in Paragraph 204, we have the following relations:

$$P = \frac{B}{2}, C = \frac{G}{7.2}, \text{A. H. P.} = F \times C$$

in which

P = required air pressure, in pounds per square inch.

B = actual submergence, in feet.

C = cubic feet of air required at pressure P .

G = gallons of water discharged per minute.

A. H. P. = air horse power.

F = horse power factor for pressure P , (from Table XLVIII., Paragraph 204.)

$$\text{D. H. P.} = \begin{cases} \text{A. H. P.} \times 1.33 & \text{for two-stage compressor.} \\ \text{A. H. P.} \times 1.66 & \text{for single-stage compressor.} \end{cases}$$

As compressors are usually rated according to the cubic feet of free air which they will compress per minute to a given higher or terminal pressure, it is necessary in computations of this kind to reduce the volume of air required at pressure P to its equiva-

lent volume at atmospheric pressure (free air), which may be done by the use of Table LXXV.

TABLE LXXV.

RELATIVE AIR VOLUMES AT DIFFERENT PRESSURES.

Gauge pressure.	Volume of free air corresponding to 1 cubic foot of air at given pressure.	Gauge pressure.	Volume of free air corresponding to 1 cubic foot of air at given pressure.
10	1.68	110	8.48
20	2.36	120	9.17
30	3.04	130	9.84
40	3.72	140	10.52
50	4.40	150	11.20
60	5.08	160	11.88
70	5.76	170	12.56
80	6.44	180	13.24
90	7.12	190	13.92
100	7.80	200	14.60

Example.—An air lift has a submergence of 100 feet and it is desired to secure a flow of 200 gallons per minute. What should be the rated capacity of the compressor and the delivered or brake horse power for operating it?

$$P = \frac{100}{2} = 50 \text{ pounds per square inch.}$$

$$C = \frac{200}{7.2} = 28 \text{ cubic feet per minute.}$$

$$F = 0.12 \text{ (from Table XLVIII).}$$

Volume of free air corresponding to 1 cubic foot at 50 pounds pressure = 4.4 (from Table LXXV.).

Capacity of compressor = $28 \times 4.4 = 123$ cubic feet of free air per minute compressed to 50 pounds gauge pressure.

$$\text{A. H. P.} = 0.12 \times 28 = 3.36.$$

As the pressure is less than 60 pounds a single-stage compressor may be used, calling for $3.36 \times 1.66 = 5.6$ D. H. P.

Fire Pumps.

305. Types Employed.—All systems of fire protection should be provided with at least two independent water supplies, and one of these frequently requires the installation of a fire pump.

Three types of pumps are used for this purpose, which should be especially designed for the conditions under which they are to operate. These include the direct-acting steam pump, the rotary pump, and turbine pump. The National rules call for four standard sizes, as follows: 500, 750, 1,000 and 1,500 gallons per minute, which applies to all three types. Very few pumps of the larger size are made, it being considered better practice to use two 750-gallon pumps which allows a remaining unit in case of accident.

The 1,000-gallon pump is the size most frequently employed, although many 750-gallon pumps are installed. The 500-gallon size is too small for good protection except in special cases.

306. Steam Pumps.—These should be of the duplex type, as they are more certain of starting promptly after long disuse. Table LXXVI. gives working data for pumps of this type.

TABLE LXXVI.
DATA FOR STEAM FIRE PUMPS.

Capacity of pump, gallons per minute.	Steam pressure, pounds gauge.	Water pressure, pounds gauge.	Piston speed, feet per minute.	Number of 1½-inch fire streams.	Boiler horsepower.
500	40	100	140	2	100
750	45	100	140	3	120
1,000	45	100	140	4	150
1,500	50	100	160	6	200

Under ordinary conditions, with a lift less than 18 feet a priming tank with suitable connections is all that is necessary for keeping the cylinders and suction pipe filled.

When the distance from the supply to the pump is over 300 or 400 feet a steam ejector connected into the suction pipe near the pump should be provided in addition to the priming tank. Auxiliary pumps are used for off-setting the effect of leakage in the system of piping and to maintain the required pressure on the sprinkler system. These usually take their supply from the suction-pipe near the main pump and discharge into the main delivery pipe. Both fire pump and auxiliary pump are started automatically by a pre-determined drop in pressure in the delivery pipe. A fire pump of this type, together with its auxiliary equipment is shown in Fig. 149.

307. *Rotary Pumps.*—These should be designed to maintain a working pressure of 100 pounds at a speed of 250 to 300 r. p. m., according to size. Pumps of this type may be driven by direct-connected motors or turbines, or by means of friction or spur gearing. When friction gearing is used about 30 D. H. P. for each 250 gallons of water per minute will be required. With spur gearing this will be considerably less.

Practically the same priming equipment is required as for a steam pump, although the tank may be somewhat smaller.

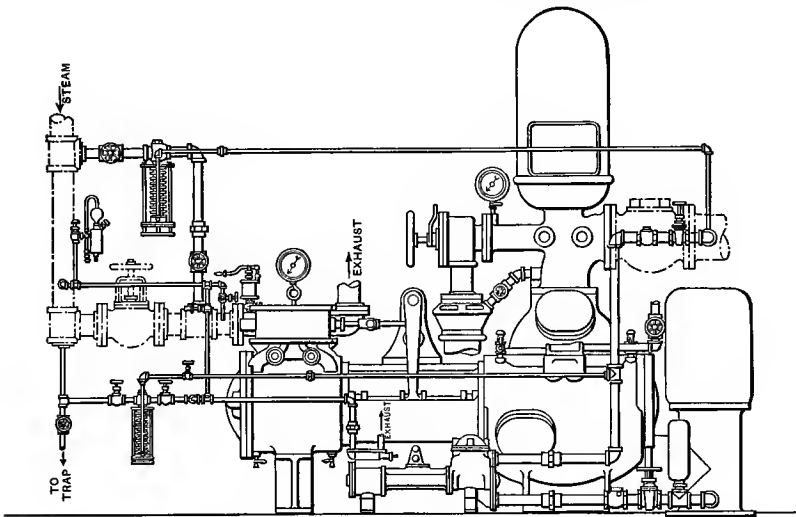


Fig. 149. Fire Pump with Auxiliary Equipment.

308. *Turbine Pumps.*—Turbine pumps used for fire purposes should be of heavy construction and have from two to four stages in order to secure the required pressure at a moderate speed. Turbine fire pumps may be driven either by motors or steam turbines, and commonly run at speeds ranging from 800 to 1,200 r. p. m. They require from 20 to 30 D. H. P. for each 250 gallons of water discharged per minute.

When possible they should take their supply under a slight head as they have no lifting ability until primed. When this is not practicable a priming tank of ample size should be provided.

When there is an especially long suction pipe, an auxiliary pump should be furnished connecting with the discharge from the main pump inside the check valve.

Detailed information for the installation of all work of this kind should be obtained from the Underwriters for the district in which the plant occurs, as previously noted.

Special Equipment.

The various steam and water specialties employed in the equipment of power and heating plants have already been described in previous volumes and reference may be made to these for more detailed information.

Boiler Feeding.

309. *Pumps Employed.*—The direct-acting steam pump is the standard type for boiler feeding, although the multiple-stage cen-

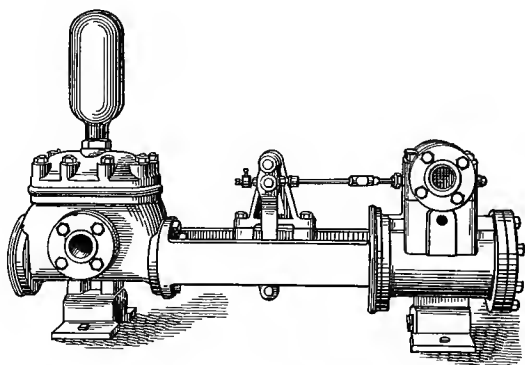


Fig. 150. Direct-Acting Feed Pump, Piston Type.

trifugal pump is used to a considerable extent in plants of large size. The triplex power pump may also be adapted to boiler feeding by making suitable provisions for regulation.

310. *Direct-Acting Feed Pumps.*—Two standard forms of direct-acting feed pumps are shown in Figs. 150 and 151. The first of these, which represents both the piston and inside plunger types, is used where the water is free from grit and where the

pump is exposed to dust or ashes, as when it is located in the boiler room.

The outside packed plunger pump (Fig. 151), is better adapted to the handling of gritty water as leaks are easily detected and adjustments made from the outside. On the other hand, the surfaces of the plungers are exposed to outside dirt which is often a disadvantage in certain locations. The speed of a direct-acting pump is easily regulated, a feature which adapts it especially to boiler feeding. When the temperature of the feed water exceeds 115 to 120 degrees it should be arranged to flow to the pump by gravity as difficulty will be experienced in raising it by suction at higher temperatures.

Feed pumps are made both single and duplex, the latter being usually preferred for this purpose as they produce a steadier flow of water to the boilers.

In determining the size of pump it is customary to allow for a theoretical capacity equal to about twice the actual capacity required by the boilers under normal conditions, which provides for leaks, slippage, overloads, etc. On this basis the boiler horse

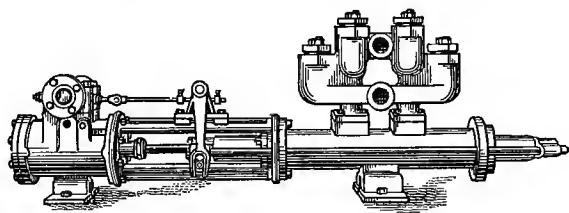


Fig. 151. Feed Pump with Outside-Packed Plunger.

power which a *single* pump cylinder will supply is given by the formula

$$\text{B. H. P.} = D \times L \times N \times 8$$

in which

B. H. P. = boiler horse-power supplied.

D = gallons delivered per inch stroke (Table LXX.).

L = length of stroke, in inches.

N = number of strokes per minute.

311. Centrifugal Feed Pumps.—The advantages of the centrifugal pump for this purpose are a steady flow of water, absence of air chambers and valves, and small liability to damage by foreign matter contained in the water. They are not usually recommended for plants requiring less than 150 to 200 gallons per minute or where there is considerable fluctuation in load. Pumps of this type may be driven either by electric motors or steam turbines (Fig. 152), the latter being the more frequent method. Regulation is best secured in large plants by dividing the work among several units which allows them to be operated at nearly full load at all times, thus obtaining the highest efficiency.

312. Triplex Power Pumps.—In order to adapt this type of pump to boiler feeding provision must be made for regulating the flow. This may be done by using a small independent steam

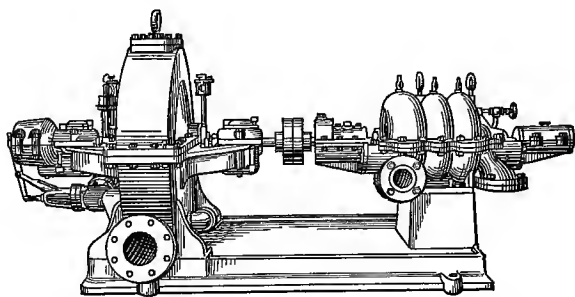


Fig. 152. Turbine-Driven Centrifugal Feed Pump.

engine geared to the pump and varying the speed, or by employing an adjustable by-pass valve when a motor drive is used. In general, any gain in steam economy over the direct-acting pump under these conditions will be offset by transmission losses.

313. Injectors.—Injectors in stationary practice, are employed principally as a reserve in case of accident to the feed pumps, except in connection with small plants or single boilers. While efficient and convenient under certain conditions, they are not adapted to isolated plants where hot water must be handled in large and varying quantities. The injector, however, has a wide field of its own, covering locomotives and portable boilers of all kinds, besides serving as a supplementary feed in stationary plants.

Feed-Water Heating.

314. Reasons for Heating Feed Water.—Feed-water heaters are employed for three reasons; first, to avoid the introduction of cold water into a boiler; second, as a means of utilizing exhaust steam which would otherwise be wasted, and third, for purifying the feed water by the process of precipitation under certain conditions. The relative importance of these will vary in different cases according to local conditions. When the exhaust is utilized for heating and ventilating purposes there is no advantage in employing a feed-water heater except in the summer when the heating system is not in use, as the small amount of water required to replace leakage losses, etc., may be mixed with the condensation returned from the heating system.

If a portion of the exhaust is wasted a heater of sufficient size should be employed to heat the "make-up" water under maximum conditions. When the feed is drawn from the hot well, exhaust from the auxiliaries may be used to raise its temperature to a point approximating that of steam at atmospheric pressure.

In condensing plants conditions are often such that it is advisable to let the condensate go to waste and utilize the exhaust in heating fresh feed water rather than attempt to remove the oil. In cases of this kind two heaters are commonly employed, one connected with the exhaust main between the engines and the condenser, and a secondary heater taking exhaust from the auxiliaries. When the exhaust from the latter source is equal to about one-sixth the entire quantity of steam generated by the boilers, there will be nothing gained by placing a low-pressure heater in the main exhaust pipe.

On the other hand, if the exhaust from the auxiliaries is less than this amount both primary and secondary heaters may usually be employed to advantage.

In general, it will be found desirable in most combined power and heating plants to install a feed-water heater for summer use and for such other times as exhaust steam may be utilized in this way.

315. Saving by Heating Feed Water.—The percentage of saving in fuel by heating the feed water with waste products is expressed by the equation

$$\frac{100(T_2 - T_1)}{L + (T_3 - T_1)}$$

in which

T_1 = initial temperature of feed water.

T_2 = final temperature of feed water.

T_3 = temperature of steam at boiler pressure.

L = latent heat of steam at boiler pressure.

316. *Types of Heaters.*—Feed-water heaters are of two general types known as *open* and *closed* heaters. In the first of these

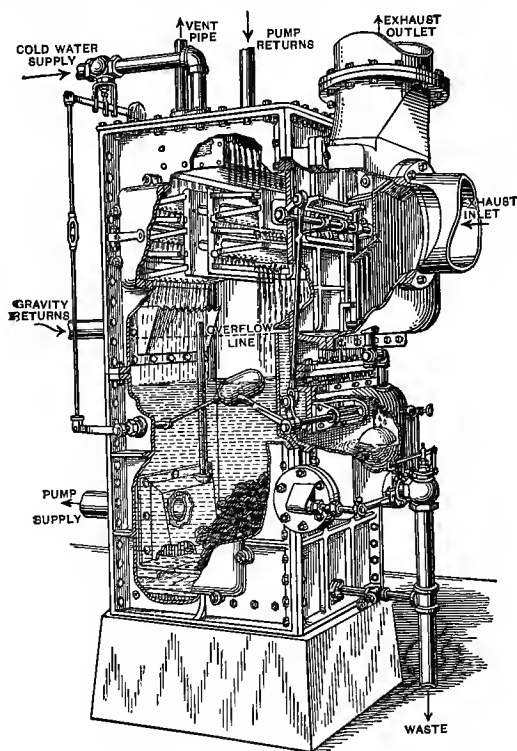


Fig. 153. Open Feed-Water Heater.

the steam and water mingle in a closed chamber. Heaters of this type are commonly employed in connection with exhaust heating plants and are made to serve as a combined heater, filter and receiver, as illustrated in Fig. 153. When properly proportioned and supplied with a sufficient amount of exhaust they are capable

of raising the temperature of the water to approximately that of the steam, in which case they are slightly more effective than the closed type. A feature of considerable importance when the feed contains salts of lime and magnesia is an arrangement of shallow trays in the upper part of the chamber where precipitation may take place before the water reaches the receiver, thus acting as a purifier as well as heater.

As the exhaust steam and water are in direct contact, the oil must be removed from the exhaust before it enters the heater.

The separator in most cases forms a part of the regular equip-

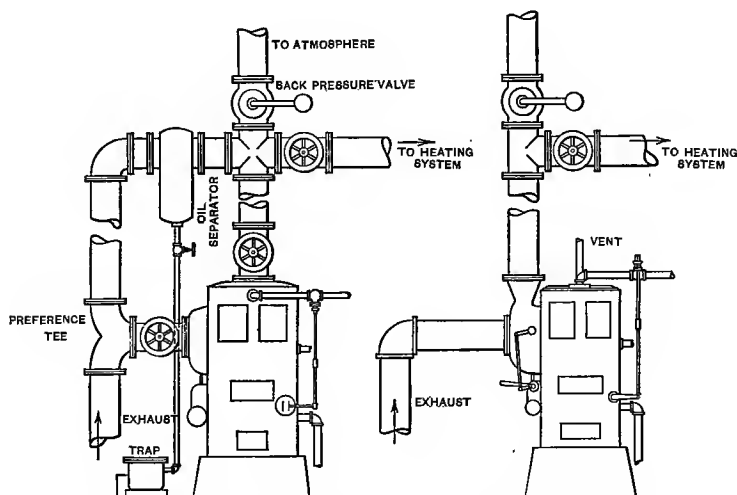


Fig. 154. Two Forms of Connection for an Open Feed-Water Heater.

ment, being bolted to the shell of the heater, as shown in Fig. 153. Two methods of making the steam connections with a heater of this type are illustrated in Fig. 154. In the arrangement shown at the left the by-pass and valves are independent of the heater, while in the second case a special cut-out valve is employed.

By this means any proportion of the exhaust may be diverted to the heater or it may be cut out entirely. The simplicity of this device is evident from the cut and the amount of piping considerably reduced. In the closed heater the steam and water are in

separate compartments, the water usually passing through a series of brass tubes surrounded by steam, although in some cases this arrangement is reversed. A water-tube heater is more efficient for a given heating surface than a steam-tube heater, owing to the higher velocity of flow, but when used as a purifier it is necessary to secure a slower movement of the water in order to obtain an effective precipitation of the solids.

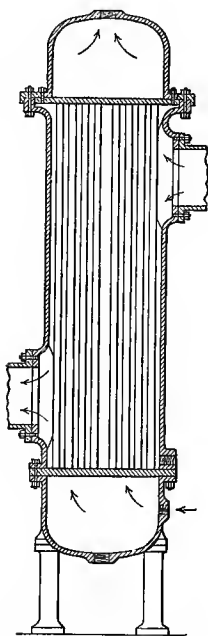


Fig. 155. Closed Feed-Water Heater, Single-Flow Type.

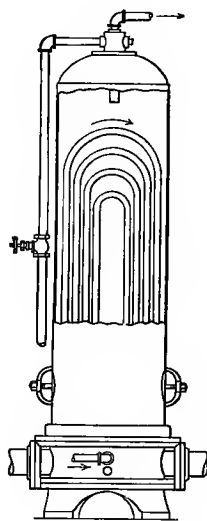


Fig. 156. Feed-Water Heater, Steam-Tube Type.

This is most easily brought about by passing the water through the shell of the heater with the steam inside the tubes.

A single-flow water-tube heater is shown in Fig. 155, the action of which is clearly indicated by the arrows. In the multi-flow type the water passes back and forth a number of times before leaving the heater.

A steam-tube heater of typical design is illustrated in Fig. 156.

Closed heaters are always placed on the pressure side of the feed pump and may therefore be located at any elevation desired.

The temperature to which the feed water may be raised with a closed heater commonly runs from 200 to 210 degrees with exhaust from non-condensing engines, and from 110 to 120 degrees in the case of condensing engines.

317. Tube Surface.—The required amount of tube surface depends upon various conditions including the steam pressure, initial and final water temperatures and velocity of flow. The form of the tube, whether straight or corrugated, and the material, also have a considerable effect. For average conditions the following approximate relations may be taken between the tube surface and quantity of water to be heated. These figures are based on an initial feed temperature of 60 degrees, with a final temperature of 208 degrees in case of non-condensing engines, and 115 degrees for condensing.

TABLE LXXVII.

TUBE SURFACE FOR CLOSED HEATERS.

Type of heater.	Pounds of water heated per hour per square foot of tube surface.	
	Non-condensing, atmospheric pressure.	Condensing, 26-inch vacuum.
Water-tube, single-flow	80	140
Water-tube, multi-flow	100	180
Steam-tube	60	100

Condensing Apparatus.

318. Conditions Governing Use of Condensers.—The advantages of condensing and the percentage of saving, or gain in power, under different conditions have been considered in Paragraph 267. Some of the more common types of condensing apparatus will now be taken up and discussed briefly in connection with the requirements of isolated plants.

In general, condensers are only made use of in factory work and plants of large size where the power requirements are fairly constant throughout the day. In city buildings where the maximum load occurs only during the lighting hours, and where the cost of cooling water would be excessive, there is usually no advantage in providing a condenser, especially if the exhaust steam

can be utilized for heating purposes during several months of the year. There are special cases, however, where cooling water may be obtained at a reasonable rate, in which it may be a matter of economy to operate the plant condensing during the summer months when the heating system is out of commission. In such cases a careful comparison should be made between the probable saving in steam consumption and the cost of maintaining and operating the condensing apparatus, including taxes, depreciation and interest on the first cost of equipment. In computations of this kind the length of time which the apparatus would be in actual use is an important factor.

In factory and mill work the conditions are more favorable for condensing, especially where steam turbines are employed. The use of the bleeder turbine makes it possible to obtain steam

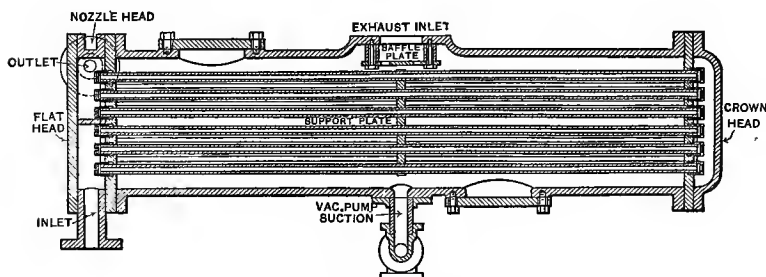


Fig. 157. Section through a Surface Condenser.

at low pressure for heating purposes without affecting the action of the condenser to any appreciable extent.

Types of Condensers.

319. General Classification.—Condensers are included in two general classes known as *surface* and *jet* condensers. The first of these is divided into the “water-cooled,” “air-cooled” and “evaporative” types, while the second is made up of the “parallel-current” and “counter-current” types. These in turn are still further sub-divided according to various details of design, but only a sufficient number will be shown to illustrate the principles of operation and to bring out some of the more important details of construction.

320. *Surface Condensers.*—The water-cooled type of surface condenser is the one commonly used in isolated plant work and is the only form which will be considered. A section through a typical surface condenser employed in connection with reciprocating engines, where the vacuum does not exceed 26 inches, is shown in Fig. 157. This is similar to a feed-water heater of the double-flow type, and requires two pumps, one for circulating the cooling water and a vacuum or air pump for removing the air and condensed steam. These pumps are often combined so as to be operated by a single steam cylinder, as illustrated in Fig. 158.

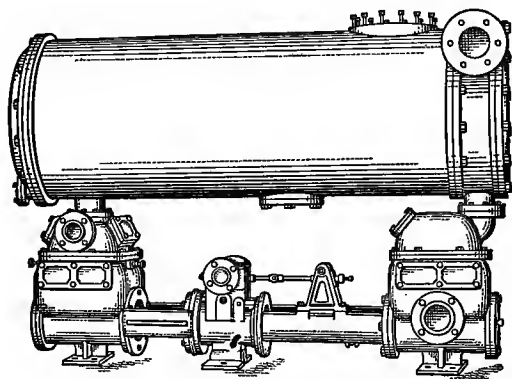


Fig. 158. Surface Condenser and Pumps.

It has been shown in the previous chapter that a higher vacuum of 28 to 29 inches is required in connection with a steam turbine to secure the highest efficiency. In order to adapt the surface condenser to this class of work it is necessary to secure a more complete removal of air and also to prevent the condensation from the upper tubes falling on those directly below as the coating of water thus formed reduces their efficiency as heating surface. This has led to the construction of the "dry-tube" condenser provided with a separate "dry-air" pump. A section through a condenser of this type showing the "rain plates," and various connections for steam, air, and condensation is illustrated in Fig. 159. Another arrangement of tubes designed to equalize the flow of steam and air in large condensers is shown in Fig. 160. A complete surface condensing outfit connected with a double-flow

turbine is illustrated in Fig. 161. The circulating pump in this case is at the left and the hot-well and air pumps at the right. All of these are of the centrifugal type driven by steam turbines. The hot-well and air pumps are mounted upon a common shaft and are illustrated in section in Fig. 162.

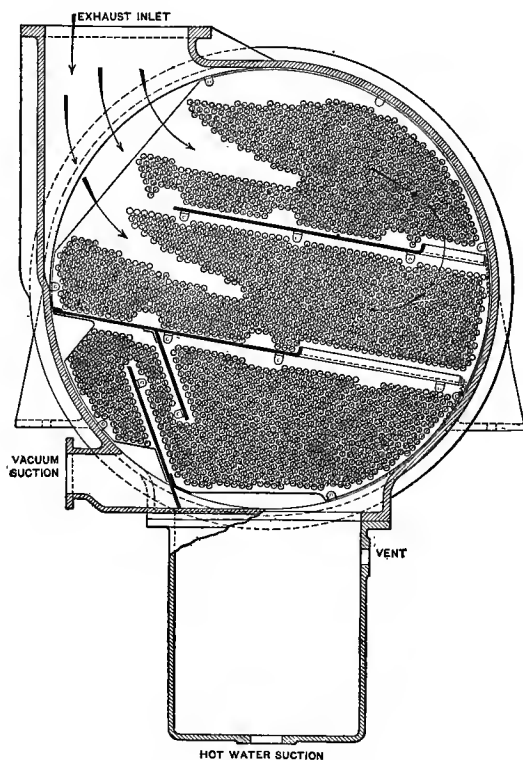


Fig. 159. Section through a Surface Condenser Showing Rain Plates.

The air pump, shown at the right, is of special design, being a combination of centrifugal pump and steam ejector through which a current of water is drawn with the air. The advantage of this over the usual type of piston air pump is its simplicity, the absence of valves and the fact that it may be attached to the same shaft with a centrifugal hot-well or condensate pump. In some cases

all three pumps are mounted upon a common shaft driven by a single turbine.

321. Tube Surface.—The required tube surface is a detail of design depending upon various factors according to the conditions under which it is to operate. As these vary more or less in different plants it is usually best to submit the given conditions to the manufacturers letting them determine the required tube surface for their particular type of condenser.

Standard condensers for reciprocating engines are commonly rated on a basis of 2 square feet of tube surface per I. H. P., which corresponds very nearly to the computed requirements for cooling water at 60 degrees, a 26-inch vacuum, and a steam consumption of 20 pounds per I. H. P. per hour.

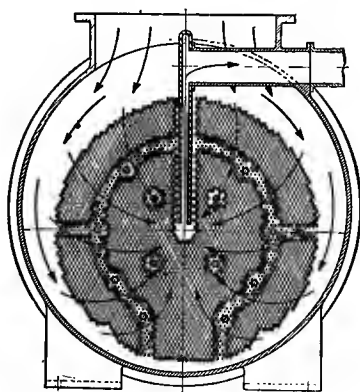


Fig. 160. Arrangement for Equalizing the Flow of Steam in Surface Condensers.

Small turbo-generators equipped with standard condensers require from 3 to 4 square feet of surface per kilowatt, and large units, employing dry-tube condensers of the best type, from 1.7 to 2.4 square feet.

The heat transmission through condenser tubes varies with the freedom from air, the condition of the surface as to dryness and cleanliness, and the velocity of both the steam and water. Under ordinary working conditions the transmission per degree difference between the steam and average temperature of the water will run from 250 to 350 B. T. U. per square foot of surface per

hour in the *standard* types. With *dry-tube* condensers having separate air pumps, these figures may usually be doubled, although some engineers recommend limiting the transmission factor to 500 B. T. U. for continuous use under the conditions of ordinary service. The formula for computing the cooling surface for a given set of conditions is

$$S = \frac{w(H - T_3 + 32)}{U \left(T - \frac{T_2 + T_1}{2} \right)}$$

in which

S = square feet of tube surface.

w = weight of steam to be condensed per hour, in pounds.

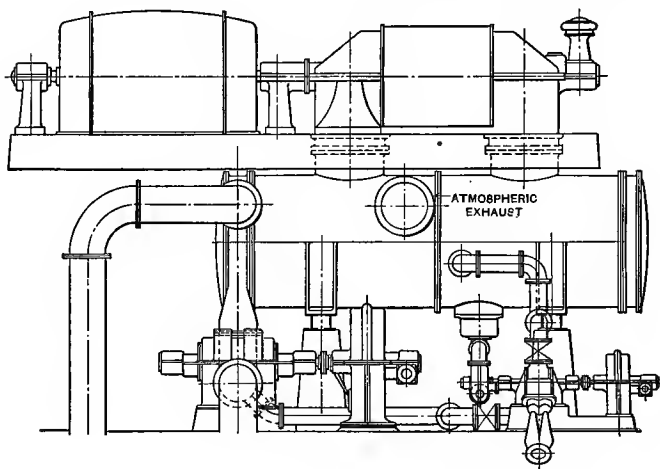


Fig. 161. Condensing Equipment Attached to a Steam Turbine.

H = total heat of steam, at condenser pressure.

U = factor for heat transmission per degree difference per square foot of surface per hour.

T = temperature of steam, at condenser pressure.

T_1 = initial temperature of cooling water.

T_2 = final temperature of cooling water.

T_3 = final temperature of the condensed steam.

In using this formula the value of U will range from about 250 to 500 according to the type of condenser as previously stated.

The final temperature of the cooling water (T_2) and that of the condensed steam (T_3) will depend upon the type of condenser and the conditions of operation.

In the case of *standard* condensers, T_2 may be taken from 15 to 20 degrees below T , and T_3 from 8 to 12 degrees below.

With the *dry-tube multi-flow* type, T_2 may be taken from 5 to 10 degrees below T , and T_3 from 4 to 8 degrees below.

322. Jet Condensers.—Two types of the jet condenser are shown, those employing pumps for the removal of air and water from the condensing chamber and those in which no pumps are required (except for supplying the cooling water in some cases), called barometric or siphon condensers.

A section through the condensing chamber of a low-vacuum counter-current condenser of the first type is shown in Fig. 163. The exhaust steam and cooling water enter at the points indicated, the latter being discharged in the form of a spray and mingling

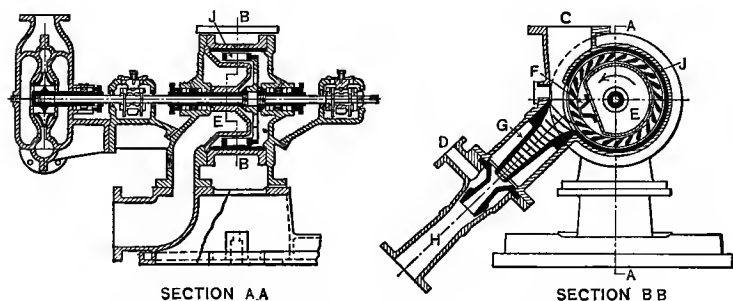


Fig. 162. Special Centrifugal Air Pump for Condenser Work.

with the entering steam. The mixture of discharge water, condensed steam and air is withdrawn by means of a single pump.

The vacuum breaker shown at the right is for the purpose of admitting atmospheric pressure to the condenser in case of accident to the discharge pump, thus breaking the vacuum and preventing the flooding of engine or turbine.

Parallel-current condensers are similar in construction to the above except the spray cone is reversed so as to throw the water downward. A jet condenser of this general type provided with a separate dry-air pump is shown in Fig. 164. Another form of

counter-current jet condenser adapted to high-vacuum work is illustrated in Fig. 165. This requires a tail pump for removing the discharge water and condensate and a separate dry-air pump. The various connections and the passages taken by the air and steam are indicated in the cut.

A form of jet condenser especially adapted to low basements is illustrated in Fig. 166. In this case the exhaust steam enters the

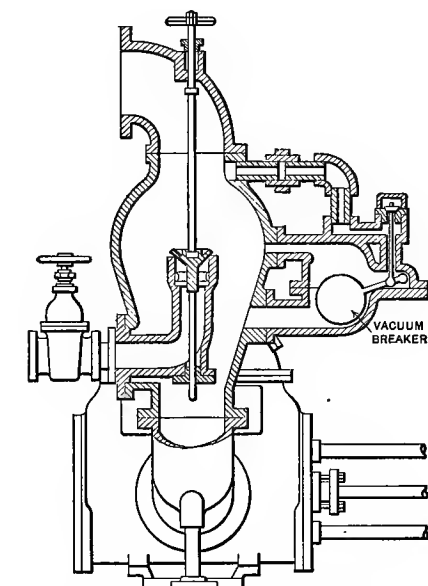


Fig. 163. Low-Vacuum Jet Condenser.

lower part of the chamber at the left and is met by finely divided streams of water flowing downward over a series of trays.

A condenser of this type connected with a steam turbine is shown in Fig. 167, which illustrates the method of removing the air and discharge water from the condensing chamber. In this case duplicate turbine-driven centrifugal pumps are employed for the latter purpose and a rotative dry-vacuum pump for the former. This arrangement of pumps is frequently employed in connection with high-vacuum surface condensers also.

A section through the condensing head of a barometric condenser of the siphon type is shown in Fig. 168 and the general method of connecting it with a reciprocating engine in Fig. 169. This is a parallel-current condenser, the cooling water entering at the right (Fig. 168), and the exhaust steam through the goose neck at the top as indicated by the arrows. Condensation takes place in the nozzle *A*, the water entering at the upper edge in the form of a hollow cone. The discharge water, together with the entrained air, passes through the contracted throat *B*, into the tail pipe and thence to the hot-well and overflow as indicated in Fig. 169.

In order to work properly the distance from the surface of the hot-well to the condensing nozzle should not be less than 34 feet, and in practice it is usual to increase this somewhat to prevent

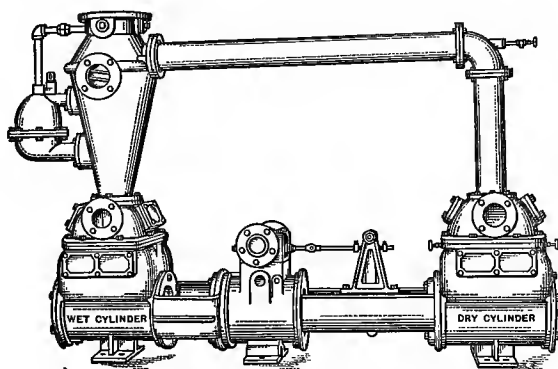


Fig. 164. Jet Condenser and Pumps.

any possibility of a reversal of flow. When the distance from the cooling-water supply to the condenser inlet does not exceed 15 feet, a flow will be established and maintained by the siphon action produced. For greater elevations a pump is necessary.

Quantity of Cooling Water.

323. Method of Computing.—An approximate method has already been given for estimating the quantity of cooling water required when computing the total supply for the plant.

The theoretical quantity of cooling water for a *jet* condenser is given by the formula

$$W = \frac{w(H - T_2 + 32)}{T_2 - T_1}$$

in which

W = pounds of cooling water required per hour.

w = pounds of steam to be condensed per hour.

H = total heat in one pound of steam, at condenser pressure.

T_1 = initial temperature of cooling water.

T_2 = temperature of discharge water.

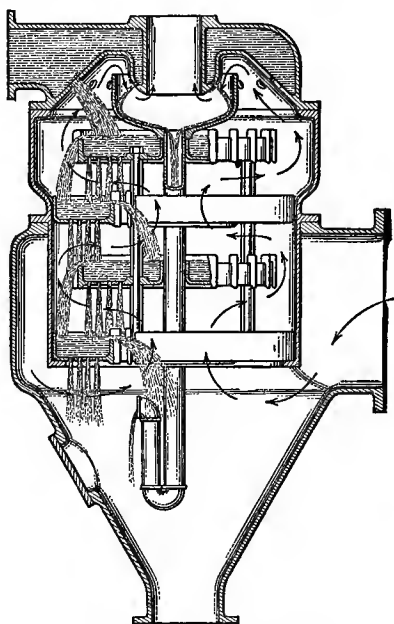


Fig. 165. High-Vacuum Jet Condenser.

In the case of reciprocating engines T_2 should be taken about 15 degrees below the temperature corresponding to the condenser pressure, or approximately 125 degrees for a 24-inch vacuum and 110 degrees for a 26-inch vacuum.

In turbine installations, T_2 will average about 100 degrees for

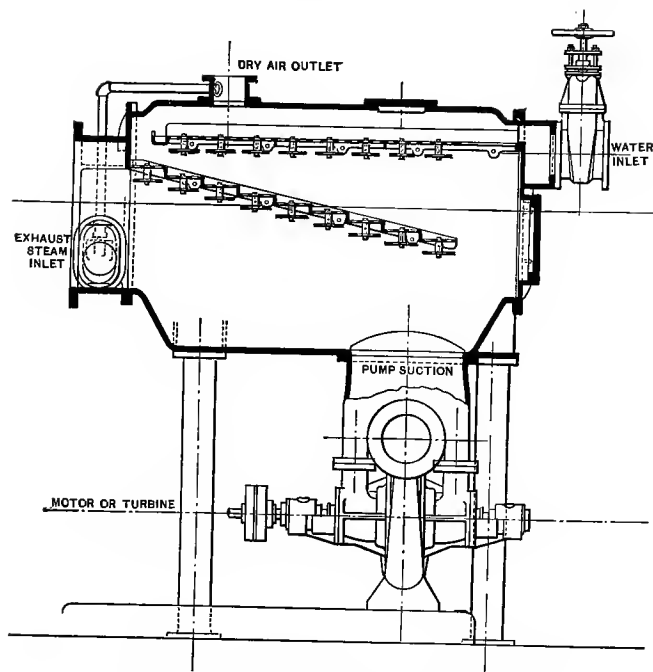


Fig. 166. Section through a Low-Head Jet Condenser.

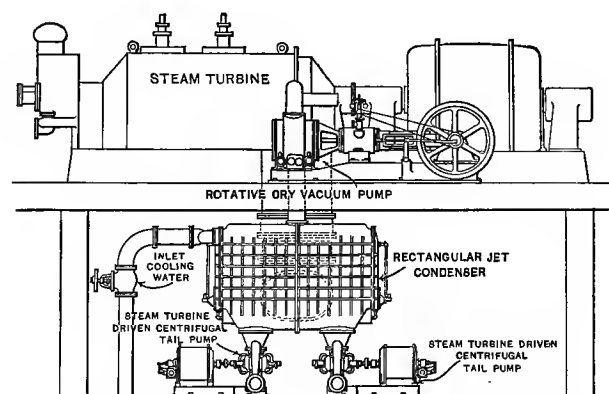


Fig. 167. Low-Head Jet Condenser and Pumps.

a 27-inch vacuum and 75 degrees for a 28.5-inch vacuum, allowing a difference of 15 degrees as before. The results given by this method should be increased from 10 to 15 per cent in actual practice, and for a *surface* condenser the amount of cooling water should be increased about 20 per cent over that required for a jet condenser under the same conditions.

Selection of a Condenser.

324. *Governing Factors.*—The best type of condenser to employ in any given case will depend upon local conditions, such as quality

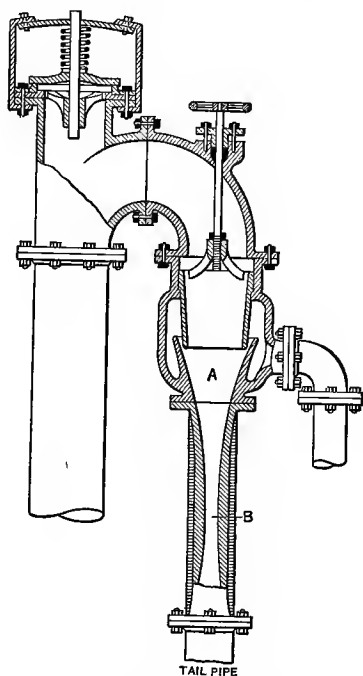


Fig. 168. Section through a Barometric Condenser.

of feed and condensing water, source of supply, available space for condensing equipment, etc.

For a low vacuum where there is an ample supply of water suitable for boiler feeding, the jet condenser with connecting

pump is a compact and satisfactory type. In mill work, where there is ample head room, the barometric type may be employed under similar conditions.

When the cooling water contains a considerable quantity of foreign matter which would tend to clog the passages of a pump or barometric condenser of the spray type, the siphon condenser (Fig. 168), may be used to advantage. In cases of this kind the boiler feed must be taken from an independent source.

Surface condensers are adapted to locations where suitable feed water is scarce, as the condensate may be used over again in the boilers by removing the oil and adding about 10 per cent of fresh water.

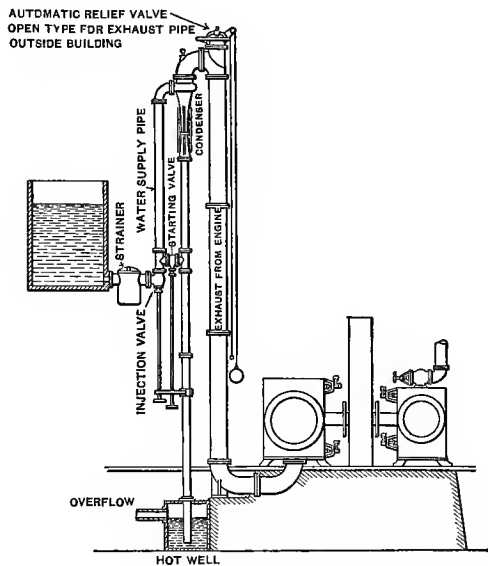


Fig. 169. Connections for a Barometric Condenser.

For the high vacuum required in connection with steam turbines the surface condenser is more commonly used although certain types of the jet condenser are now designed for this particular class of work.

CHAPTER IX.

PIPING FOR POWER PLANTS.

325. *Drawings.*—All heavy piping for power and heating work should be carefully laid out to scale in order to determine if there is sufficient room for the necessary fittings. In plants of large size it is customary to prepare separate drawings of the different systems such as high pressure, exhaust, feed and condenser piping, etc., as this simplifies the work of installation. Assembled drawings, including all of the different systems should also be made in order to avoid interference and to present the general scheme of piping as a whole.

Piping Materials.

326. *Metals Employed.*—Wrought iron, mild steel, brass and copper are the materials commonly employed in the manufacture of pipe for power and heating work.

Formerly, wrought iron was used in the best class of work, and is still preferred by some engineers, on account of its greater ductility and resistance to corrosion as compared with steel.

Improvements in the manufacture of steel have now made it practically equal to wrought iron for general piping work, and owing to its lower cost it is now extensively used both for power and heating.

What is commonly known as wrought-iron pipe is really mild steel and is usually furnished unless "puddled iron" is actually called for.

Brass is generally employed for all feed piping between the pumps and boilers on account of its resistance to corrosion and pitting.

Weight and Strength of Pipe.

327. *Classification and Use.*—Wrought-iron and steel pipe are commonly classified according to weight or thickness of material

as "Merchant" or "Standard," "Full-weight," Extra Heavy," and "Double Extra Heavy." Standard pipe is commonly used for heating work, exhaust lines, and all pressures below 100 pounds. For pressures of 100 to 200 pounds, full-weight pipe should be employed, and some engineers start with extra heavy pipe at 150 pounds.

Tests show the actual bursting pressure of mild steel pipe from 2 to 6 inches in diameter to range from 5,000 to 3,000 pounds per square inch, and it is commonly subjected to a hydrostatic pressure at the mills of 500 to 1,000 pounds. Extra heavy and double extra heavy pipe is tested to a maximum pressure of 1,500 pounds. The extra thickness of metal is employed largely for offsetting the effects of corrosion and prolonging the length of the pipe rather than for strength when new. While extra heavy pipe is often employed for the higher pressures it is not so flexible as the full-weight pipe, and for this reason the latter is usually preferred in the average plant of moderate size.

Extra heavy pipe, however, should always be used for the blow-off connections to boilers, for buried returns, and in all other cases where the pipes occupy inaccessible places or are especially exposed to corrosion.

Standard and full-weight pipe are designated by the nominal inside diameter up to and including 12 inches. Above this the outside diameter is employed.

The heavier weights of pipe have the same outside diameter as the standard, the extra thickness of metal being added to the inside, thus making it possible to use the same size of fitting for each weight of pipe.

Riveted pipe of sheet steel or boiler plate is sometimes used for low pressures where large diameters are called for. While amply strong for the pressures carried it is not so durable as standard pipe owing to the lightness of the material.

The brass pipe employed for feed and water connections is a seamless drawn tubing made in sizes to correspond with standard wrought-iron pipe.

Table LXXVIII. gives data relating to standard weight wrought-iron pipe.

TABLE LXXVIII.

DIMENSIONS OF STANDARD WROUGHT-IRON PIPE.

Nominal size, inches.	Circumference.		Cross Section Area.		Length of pipe for 1 square foot of.		Contents, 1 foot length.		Length for 1 cubic foot contents feet.
	External inches.	Internal inches.	External square inches.	Internal square inches.	External surface feet.	Internal surface feet.	Weight of water, pounds.	U. S. gallons.	
$\frac{3}{4}$	3.290	2.589	0.866	0.533	3.637	4.635	0.230	0.023	270.60
1	4.131	3.292	1.357	0.862	2.904	3.645	0.373	0.040	166.90
$1\frac{1}{4}$	5.215	4.335	2.164	1.496	2.301	2.768	0.648	0.063	96.25
$1\frac{1}{2}$	5.963	5.061	2.835	2.038	2.010	2.371	0.883	0.091	70.66
2	7.461	6.494	4.430	3.355	1.608	1.848	1.454	0.163	42.91
$2\frac{1}{2}$	9.032	7.753	6.491	4.783	1.328	1.547	2.072	0.255	30.10
3	10.996	9.636	9.621	7.388	1.091	1.245	3.202	0.367	19.50
$3\frac{1}{2}$	12.566	11.146	12.566	9.887	0.955	1.077	4.285	0.500	14.57
4	14.137	12.648	15.904	12.730	0.849	0.949	5.517	0.642	11.31
5	17.477	15.849	24.299	19.990	0.687	0.757	8.668	1.020	7.20
6	20.813	19.054	34.471	28.889	0.577	0.630	12.521	1.460	4.98
7	23.955	22.063	45.663	38.737	0.501	0.544	16.790	2.000	3.72
8	27.096	25.076	58.426	50.039	0.443	0.478	21.688	2.610	2.88
9	30.238	28.076	73.715	63.633	0.397	0.427	27.580	3.300	2.29
10	33.772	31.477	90.762	78.838	0.355	0.382	34.171	4.080	1.82
12	40.055	37.700	127.677	113.098	0.299	0.319	49.017	5.870	1.27
14	47.124	44.768	176.715	150.485	0.255	0.268			0.90
16	53.410	51.520	226.980	211.240	0.225	0.233			0.68

Fittings.

328. *Material and Use.*—The fittings used for making up wrought-iron pipe are usually of cast iron, although cast and pressed steel are used to some extent for very high pressures and superheat.

Cast-iron fittings are regularly made in two weights; standard for pressures up to 100 or 125 pounds and extra heavy for higher pressures not exceeding 250 pounds. It should be borne in mind, however, that fittings usually fail from other stresses than those due to internal pressure, and the elimination of excessive expansion strains, and pockets for the accumulation of condensation, are of equal importance with the weight of fitting used.

Flanged Joints.

329. *Types Employed.*—The matter of pipe joints is an important one in high-pressure work and should receive careful attention.

While screwed couplings may be employed for permanent work it is necessary to provide a large number of removable sections made up with joints which may be easily connected and discon-

nected both in erection and removal for repairs. Screwed brass unions are commonly used for pipes 2 inches and less in diameter and flanges for larger sizes.

For pressures below 100 pounds the pipe may be simply screwed into the flange as shown in Figs. 170 and 171. In the first of these the flanges are plane and in the second recessed in order to prevent the packing from being blown out. While the latter arrangement makes a tight joint it is necessary to spring it apart when removing a section for repairs, which is often a difficult matter in the case of heavy piping.

With higher pressures the points are apt to leak around the threads and other methods for attaching the flange to the pipe must be employed. One of the simplest of these is shown in Fig.

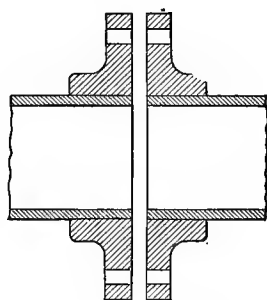


Fig. 170.

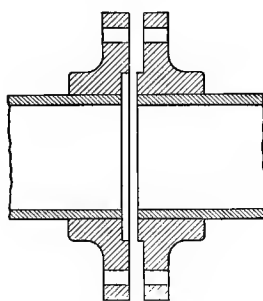


Fig. 171.

Fig. 170. Flanges with Plane Faces.

Fig. 171. Recessed Flanges.

172 in which the inner edge of the flange is rounded and the end of the pipe peened as indicated in the cut. The joint illustrated in Fig. 173 is made by boring the flanges for a shrink fit and forcing them over the pipe at a red heat. After cooling, the end of the pipe is beaded over or flared into a taper recess in the flange.

For sizes 4 inches and above the "Van Stone" or "lap joint," shown in Fig. 174, is extensively used. The pipe is flanged as indicated and faced off to give a true bearing. The flanges are loose on the pipe and are drawn up by means of bolts in the usual manner.

The most satisfactory, and also the most costly method, is to weld the flange to the pipe as indicated in Fig. 175. In joints of

this kind the faces of the flanges are usually raised slightly inside the bolt circle and ground to a steam-tight fit.

The designs shown in both Figs. 174 and 175 are adapted to high-pressure and high-temperature work although the latter is considered the standard. The gaskets used in connection with flanged joints should be selected with reference to the pressure and temperature of the steam. There are many varieties upon the market and a material adapted to the case in hand may be obtained from any reliable manufacturer.

Valves.

330. *Gate Valves* are generally employed in all steam lines as they offer a minimum resistance, and when properly placed, do

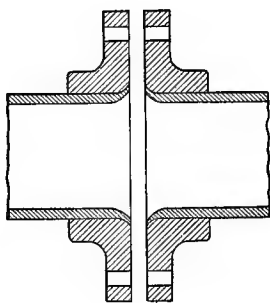


Fig. 172.

Fig. 172. End of Pipe Peened over Flange.

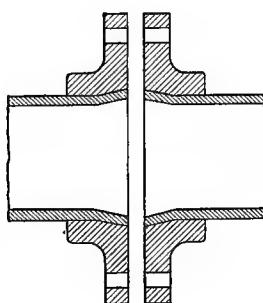


Fig. 173.

Fig. 173. Flange Shrunk in Place and Pipe Flared.

not form pockets for the accumulation of condensation and scale. They should never be set, however, with the spindle in a vertical position and the hand-wheel downward, as an obstruction to the flow of water at the bottom of the pipe is formed if only partly opened.

331. *Globe Valves* are used principally in connection with water piping and in vertical steam pipes, except the form known as an *angle valve*, which may often be employed to take the place of an elbow fitting in addition to its use as a stop valve.

Globe valves of whatever form should always be set to close against the pressure, and when used in horizontal steam pipes the

straightway valve should be placed with the stem in a horizontal position.

332. Points to be Considered.—Valves 3 inches in size and smaller are usually made with screw tops, while larger ones are provided with bolts and flanges. For high-pressure work the outside yoke and screw pattern is preferable, as the engineer is able to tell at once the position of the gate, and for this reason they are commonly employed for valves 4 inches in size and over. In the case of exhaust piping in isolated plants the inside screw valve is more frequently used on account of the limited space required.

High-pressure valves 8 inches in size and larger, should in general, be of the by-pass type. Valve bodies are usually made of brass for sizes up to and including $2\frac{1}{2}$ inches, and of cast iron

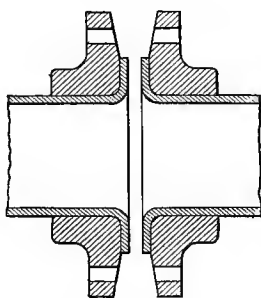


Fig. 174.

Fig. 174. Van Stone or Lap Joint.

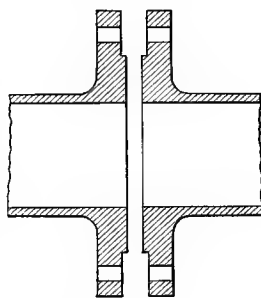


Fig. 175.

Fig. 175. Flanges Welded to the Pipe.

and steel for the larger sizes, the latter material being adapted especially to high temperatures and pressures.

Valve seats and disks should always be of a material adapted to the temperature of the steam. Babbitt metal is commonly employed for low pressures, bronze for high pressures, and nickel for superheated steam.

333. Special Valves.—When several high-pressure boilers are connected with a common header, automatic non-return valves are often provided at the outlet nozzle of each in place of the ordinary gate or angle valve. These are designed to act as a combined stop and check valve, and to close automatically in case of a sudden rush of steam from the header to the boiler, such as might occur

with the blowing out of a tube. Valves of this type also serve to equalize the pressure between the different units of a battery.

In large power plants *emergency* valves are usually placed in the various high-pressure mains and arranged to be operated from a distance. When used in the lead to an engine or turbine they are generally controlled by a special governing device which closes them upon a certain increase in speed. Pressure reducing valves, back-pressure valves, safety-valves, blow-off valves, and checks have been fully described in a previous volume and will not be considered here.

Pipe Expansion.

334. Amount of Expansion.—The expansion of wrought-iron and steel pipe under different conditions, may be determined by the formula

$$l = 0.00009(T_1 - T_2)L$$

in which

l = increase in length, in *inches*.

T_1 = temperature of the steam in the pipe.

T_2 = temperature at which the pipe was fitted.

L = original length of pipe, in *feet*.

Example.—A steam main 150 feet long is erected at a temperature of 50 degrees. How much will it increase in length when filled with steam at 150 pounds gauge pressure?

$T_1 = 366$ (from Table LXII.).

$T_2 = 50$.

$L = 150$.

which substituted in the formula gives

$$l = 0.00009 \times (366 - 50) \times 150 = 4.3 \text{ inches.}$$

335. Providing for Expansion.—The method of providing for the expansion of steam piping will depend largely upon local conditions. The use of expansion joints and the location of anchors for long runs of horizontal piping have already been discussed in Paragraphs 173 to 176 in connection with central heating. In isolated plants, the runs of piping are of comparatively short length especially those between boilers and engines, and the expansion in high-pressure systems is largely cared for by means of offsets and

long radius bends, examples of which are illustrated in Figs. 176 to 183. The first of these is the simplest method employed for taking up the expansion in a straight run of pipe where there is ample space between centers. For limited spaces the form shown in Fig. 177 may be used.

For pipes of larger size, greater flexibility may be secured by employing manifolds connected by a number of small bends, the

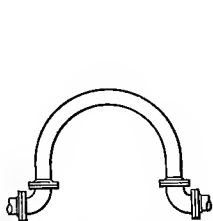


Fig. 176.

Fig. 176. Standard Expansion Loop.

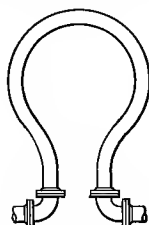


Fig. 177.

Fig. 177. Special Form of Expansion Loop.

total capacity of which is equal to that of the main pipe. An arrangement of this kind is illustrated in Fig. 178.

When there is not sufficient space for a spring offset, a double-swing, or swivel joint is often used in which the movement is

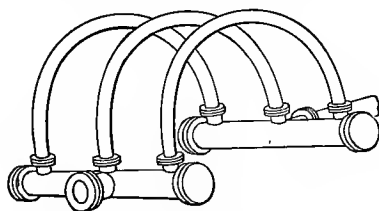


Fig. 178. Multiple Expansion Loop.

taken up by the turning of the short vertical pipes within the fittings. This method is only adapted to pipes below 8 or 9 inches in diameter, and where the movement is not excessive, as the continual turning of the pipe tends to wear the threads and loosen the joints. This arrangement is confined largely to heating and other low-pressure work.

Different methods of making the connections between the

boiler nozzles and main headers by means of sweep bends are shown in Figs. 179, 180 and 181.

When space is limited and the movements due to expansion are small, cast-iron fittings may be employed as in Fig. 182.

An approved method of connecting two headers is shown in Fig. 183. This makes use of a U-bend with valves at each end to avoid the formation of a pocket.

336. Design of Pipe Bends and Offsets.—The radius of a pipe bend should never be less than five diameters and a length of straight pipe equal to two or three diameters should be provided at each end for handling in the process of bending and for the attachment of guides or other supports.

The required length of pipe in a bend or offset to take up a given amount of expansion is given in Table LXXIX., which is

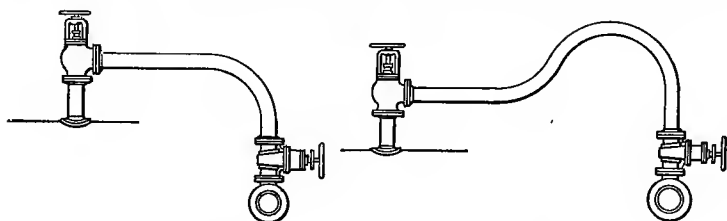


Fig. 179.

Fig. 180.

Fig. 179. Common Form of Boiler Lead.

Fig. 180. A More Flexible Form of Lead.

based upon a maximum fiber stress of 12,000 pounds per square inch in the pipe.

In using the table it should be remembered that the point of failure in work of this kind is at the joints rather than in the pipe itself and it is best in practice not to exceed about one-half the quantities given in Column 1 for the corresponding offsets, unless great care is taken to remove the strain from the fittings by means of anchors and guides. In any case extra heavy fittings should be used, and for high-pressure work only the welded or Van Stone flange should be employed under these conditions.

The length of pipe in the expansion piece may be the same whether in the form of a single right-angle offset, a double offset or U-bend.

TABLE LXXIX.

LENGTH OF EXPANSION OFFSETS AND BENDS.

Total expansion in inches.	Feet of pipe in offset at U bend for different diameters of pipe.									
	2 inches.	3 inches.	4 inches.	5 inches.	6 inches.	8 inches.	10 inches.	12 inches.	14 inches.	16 inches.
1	11	13	15	17	19	21	23	25	27	30
2	15	18	21	23	26	29	32	35	38	42
3	18	22	26	29	32	36	40	43	48	52
4	21	26	30	34	37	42	47	50	56	58
5	24	30	34	38	41	47	53	57	63	65
6	27	33	37	41	45	52	58	63	69	71
7	30	36	40	44	48	56	62	68	74	
8	32	39	43	47	52	60	66	72		

337. *Method of Reducing Stress.*—In order to reduce the stress in piping it is customary to install the expansion pieces cold and spring them in such a direction that when filled with steam, and raised to a high temperature, the final position will approximate that of the normal.

338. *Anchors.*—All anchors should be rigid to secure the best results and where the strain is always in the same direction the pipe may be solidly fastened to a beam or pier. Figs. 184 to 187, inclusive, show various methods of anchoring and supporting the

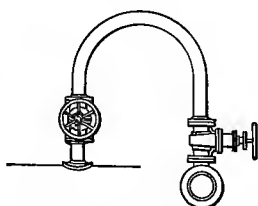


Fig. 181. Typical Connection between Boiler and Main Header.

mains in fire-proof buildings and the corresponding locations of expansion joints when these are used. Forms of anchors and guides used in connection with slip and corrugated joints are shown in Fig. 188.

Pipe Supports.

339. *Forms Employed Under Different Conditions.*—Steam and exhaust pipes should not only be strongly supported, but pro-

vision must also be made to allow for the movement due to expansion and contraction.

Supports are classified as *hangers*, *wall brackets*, and *floor stands*.

Hangers are commonly attached to the overhead construction, clamps being used for connecting with iron beams while plates or bars are embedded in reinforced concrete to which the hangers may be attached. All hangers should be adjustable in order to secure proper alignment of the piping and should be free to swing with the expansion.

Three types of wall supports are shown in Figs. 189, 190 and 191. In the first of these the pipe rests in a cradle or guide and is free to slide in either direction with the expansion and con-

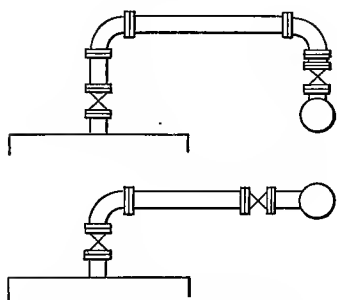


Fig. 182. Boiler Connections for Limited Head Room.

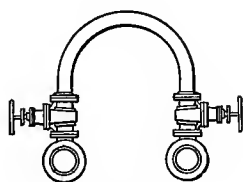


Fig. 183. Method of Connecting Two Headers.

traction. For larger pipes a roller bearing is preferable as illustrated in Fig. 190. In both cases alignment is secured by the use of adjusting nuts and slots. Where it is necessary to prevent the pipe from buckling under expansion an arrangement similar to that in Fig. 191 may be used in which the pipe is held in position by means of an upper roll and a pair of spring rods or supports. Typical methods of anchoring pipes to wall brackets are shown in Figs. 192 and 193.

In the case of floor stands the construction is practically the same as above described except the rolls or cradles are supported upon cast-iron standards or concrete piers instead of wall brackets, (see Fig. 184).

Pipe Covering.

340. *General Requirements.*—All steam and hot-water piping should be insulated with sectional covering of a material adapted to the temperature of the pipe. For high pressure pipes over 4 inches in diameter the covering should be from 1½ to 2 inches in thickness according to size. For smaller pipes and low-pressure work 1 inch is usually sufficient.

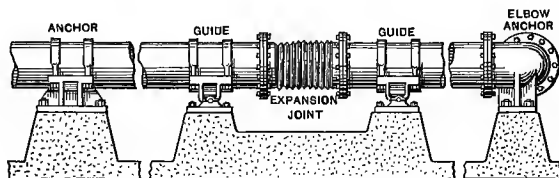


Fig. 184. Pipe Main Supported on Piers.

Boiler drums, heaters, tanks, smoke pipes, etc., should be insulated with a good form of block covering finished with plastic cement, having a total thickness of at least 2 inches.

Wherever insulation of any kind is liable to mechanical injury it should be provided with an extra covering of heavy canvas.

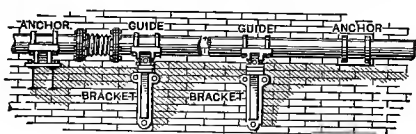


Fig. 185. Pipe Supported on Wall.

The saving effected by the use of a suitable commercial covering properly applied will amount to 75 to 90 per cent of the heat loss from the bare pipes.

Pipe Sizes.

341. *Method of Computing.*—While the pipe connections between boilers and engines are often proportioned according to the size of outlets furnished by the builders, this method should only be followed to a limited extent as it gives sizes considerably larger than are ordinarily required and thus adds unnecessarily to the cost of the plant.

For ordinary conditions satisfactory results may be obtained by assuming a velocity of flow based upon actual experience and proportioning the pipes according to the weight of steam they are to carry in a given time.

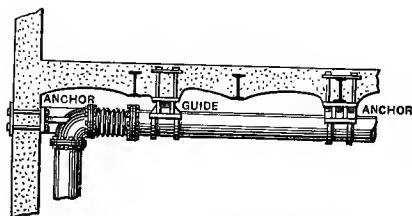


Fig. 186. Pipe Main Hung from Ceiling.

The velocities used depend primarily upon the allowable drop in pressure, which is commonly taken from 1 to 2 pounds for each

100 feet length in the case of *saturated* steam.

With straight runs of pipe, velocities of 5,000 to 10,000 feet per minute for diameters of 4 to 12 inches will give results within the above limits, but on account of bends, valves and other obstructions found in actual piping systems, it is well to reduce the above velocities about 20 per cent, making the range from 4,000 to 8,000 feet per minute.

Velocities for intermediate pipe sizes may be proportioned according to their internal areas by use of Table LXXVIII.

For *superheated* steam, velocities of 5,000 to 10,000 feet may be used in place of the above.

In case of *exhaust* pipes the velocity should be kept low in order to reduce the back pressure, and velocities from 2,000 to 4,000 feet per minute may be used for pipes ranging from 5 to

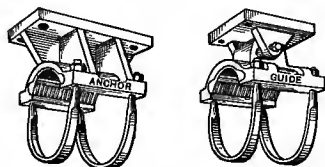


Fig. 188. Details of Anchor and Guide.

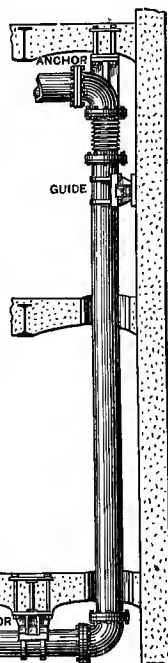


Fig. 187. Method of Supporting a Large Riser.

15 inches in diameter for average conditions.

When the mains are much over 200 feet in length it is well to

check the sizes computed as above by the methods given in Paragraph 171.

When based on the velocity of flow the formula

$$A = \frac{144 \times W \times S}{V}$$

may be used, in which

A = internal area of pipe in square inches, (see Table LXXVIII.).

W = pounds of steam flowing through pipe per minute.

S = volume of 1 pound of steam, in cubic feet, at given pressure and temperature, (see Table LXXX.).

V = velocity of steam, in feet per minute.

TABLE LXXX.

SPECIFIC VOLUME OF STEAM.

Gauge pressure, pounds per square inch.	Volume of 1 pound of steam, in cubic feet, (S).	
	Saturated.	Superheated 150 degrees.
80	4.6	5.7
100	3.8	4.7
120	3.3	4.1
140	2.9	3.6
160	2.6	3.2
180	2.3	2.9
200	2.1	2.6

Example.—What size steam pipe will be required to supply 300 pounds of steam per minute at 180 pounds pressure and 150 degrees superheat, allowing a velocity of 6,000 feet per minute?

$W = 300$.

$S = 2.9$, (from Table LXXX.).

$V = 6,000$,

which substituted in the formula gives

$$A = \frac{144 \times 300 \times 2.9}{6,000} = 20.8 \text{ square inches.}$$

Referring to Table LXXVIII. this is found to correspond closely to a 5-inch pipe.

Steam Leads to Turbines, Pumps, and Engines.

342. *Method of Determining Size.*—In the case of turbines, and direct-acting pumps taking steam at full stroke, the flow through the supply pipes or leads is practically continuous, and the weight to be passed through the pipe per minute is given by the equation

$$W = \frac{\text{D. H. P.} \times \text{W. R.}}{60}$$

in which

W = pounds of steam per *minute*.

D. H. P. = delivered or brake horse power.

W. R. = water-rate, in pounds of steam per D. H. P. per hour.

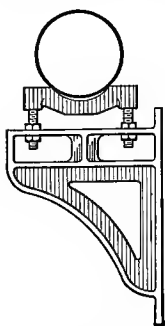


Fig. 189.

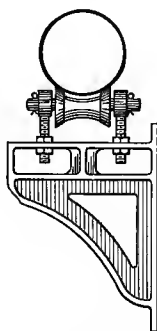


Fig. 190.

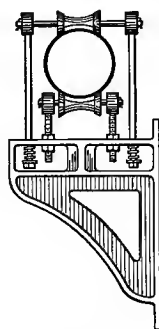


Fig. 191.

Fig. 189. Wall Support for a Medium Size Pipe.

Fig. 190. Wall Support for a Large Pipe.

Fig. 191. Pipe Support with Upper Roll.

With reciprocating engines, cutting off at a fraction of the stroke the conditions are different, as the total amount of steam supplied must flow through the pipe during the admission periods only. That is, for one-fourth cut-off the entire volume of steam supplied per minute must actually pass through the pipe in one-fourth of a minute. With a receiver separator of good size close to the throttle we may assume the total flow to take place in three-fourths of a minute. Therefore, for reciprocating engines we have

$$W = \frac{\text{D. H. P.} \times \text{W. R.}}{60} \times r$$

in which

r = the ratio of expansion, where the supply pipe leads directly from the main header to the cylinder, and

$$W = \frac{\text{D. H. P.} \times \text{W. R.}}{60} \times \frac{4}{3}$$

when a receiver separator, having a volume at least three times that of the cylinder, is placed close to the throttle.

Example.—A 100-horse-power engine, without a receiver, cutting off at one-third stroke, requires 30 pounds of saturated steam per D. H. P. per hour. Initial pressure 120 pounds per square inch.

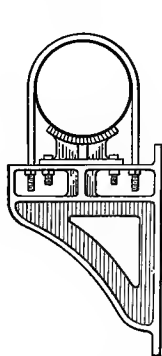


Fig. 192.

Fig. 192. Strap Anchor.

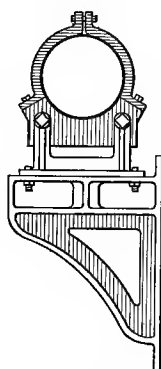


Fig. 193.

Fig. 193. Special Clamp Anchor.

What diameter of steam pipe will be required for a velocity of approximately 5,000 feet per minute?

$$W = \frac{100 \times 30}{60} \times 3 = 150.$$

$$A = \frac{144 \times 150 \times 3.3}{5,000} = 14.2 \text{ square inches,}$$

which is slightly more than the area of a 4-inch pipe.

343. Receiver Separators.—Mention has already been made of placing a receiver separator in the supply pipe close to the cylinder. The advantage of this arrangement is to provide a means of

cushioning the steam near the throttle, thus preventing vibration in the piping system and producing a uniform flow of steam from the boiler to the engine. To be effective the separator should have a volume at least three to four times that of the cylinder. With these proportions the main header may be reduced to a size just sufficient to carry the steam at the desired velocity, while the leads from the header to the separators may be made from one to two sizes smaller than called for by the engine builders, keeping the connection between the throttle and cylinder, however, full size.

344. General Considerations.—In determining the size of any given pipe for whatever purpose the maximum requirements should be taken into account. For example, in the case of boiler leads an allowance for 30 to 50 per cent overload should be provided, and so on for other conditions.

The required size of main header at the boilers will depend upon the provisions made for the removal of moisture from the steam. When large and efficient separators are furnished near the throttle valves the size of header may be based upon the flow of steam the same as any other main without regard to the matter of drainage.

Systems of High Pressure Piping.

345. General Rules for Design.—While it is customary in central stations to lay out the piping systems along general lines, it is not possible to do this to any great extent in isolated plants, except perhaps, in the case of factories and institutions where separate power houses are provided. There are, however, certain general rules which should be followed in any case and which may be readily adapted to the conditions ordinarily found in this class of work. Among these may be mentioned the following:

- 1.—The steam should be carried from the boilers to the engines in as direct a manner as possible under the existing conditions.

- 2.—The piping should be well supported and anchored to prevent vibration, while at the same time provision must be made to care for the movements due to expansion and contraction.

- 3.—All piping should be made steam and water-tight by the use of joints and packing adapted to the special conditions for which it is used.

4.—Radiation losses from pipes and other heated surfaces should be reduced to a minimum by the use of suitable insulation.

5.—Supply piping should be so proportioned as to deliver the required amount of steam without too great a drop in pressure. At the same time the velocities in large plants should be increased

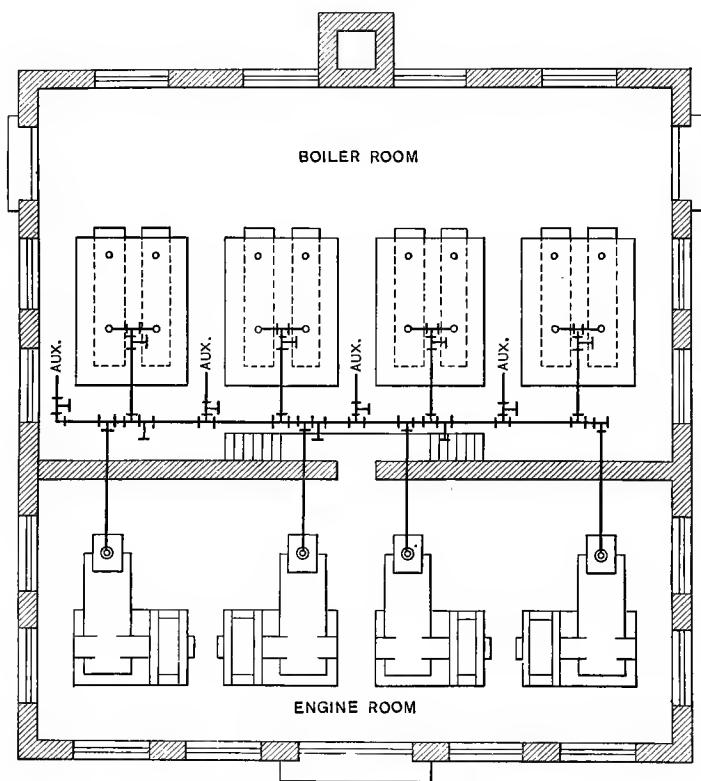


Fig. 194. Single Header System of Piping.

sufficiently to avoid the use of excessively large pipes. Low velocities should be maintained in exhaust lines in order to reduce the back pressure upon the engines to a minimum.

6.—The system should be divided into sections, or duplicated sufficiently to avoid serious delay in case of repairs. Separate

headers for engines and auxiliaries are recommended in plants of medium and large size.

7.—The drainage system is an important detail and special care should be given to the removal of condensation from all separators, headers; or low points in the piping which cannot be avoided.

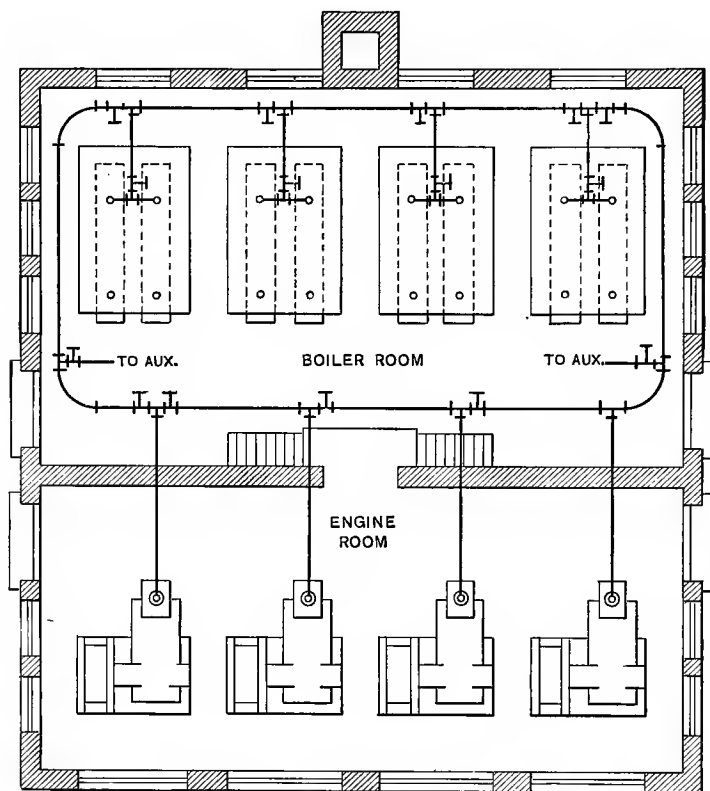


Fig. 195. Single Loop System of Piping.

Four different arrangements of high-pressure piping which may be adapted to isolated plant work are shown in Figs. 194 to 197.

346. *Single-Header System*.—The single-header system, illustrated in Fig. 194, is one of the simplest and most frequently used

in this class of work. When valved as shown in the diagram each pair of units is independent and may be cut out without interfering with the remainder of the plant. It will be noted that each section

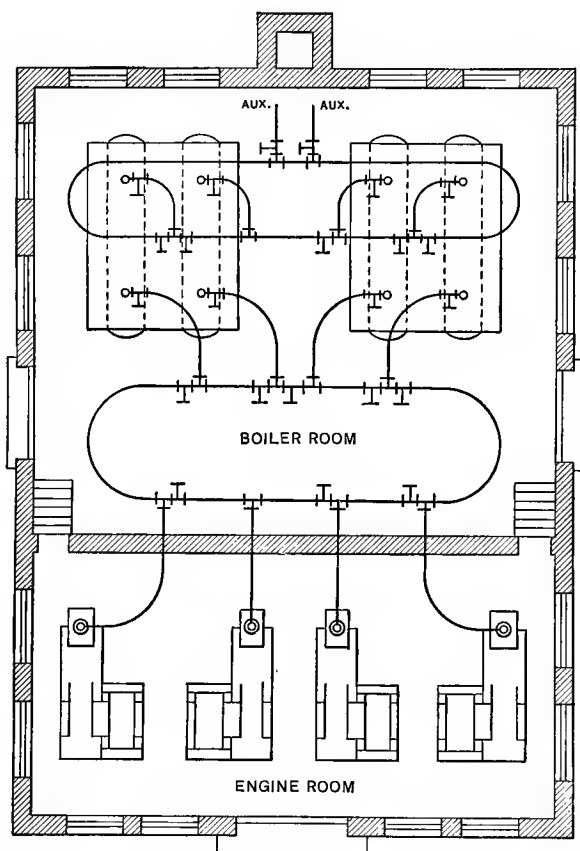


Fig. 196. Double Loop System of Piping.

of the header has a separately valved connection leading to the auxiliaries.

347. Single-Loop System.—One form of the loop or ring system is shown in Fig. 195. This is especially adapted to office buildings where there are several engines and auxiliaries to be

supplied. The main is so valved that any single unit or any number of adjacent units may be cut out independently of the others.

348. *Double-Loop System*.—This is similar to the above except an independent header is provided for the auxiliaries (see Fig.

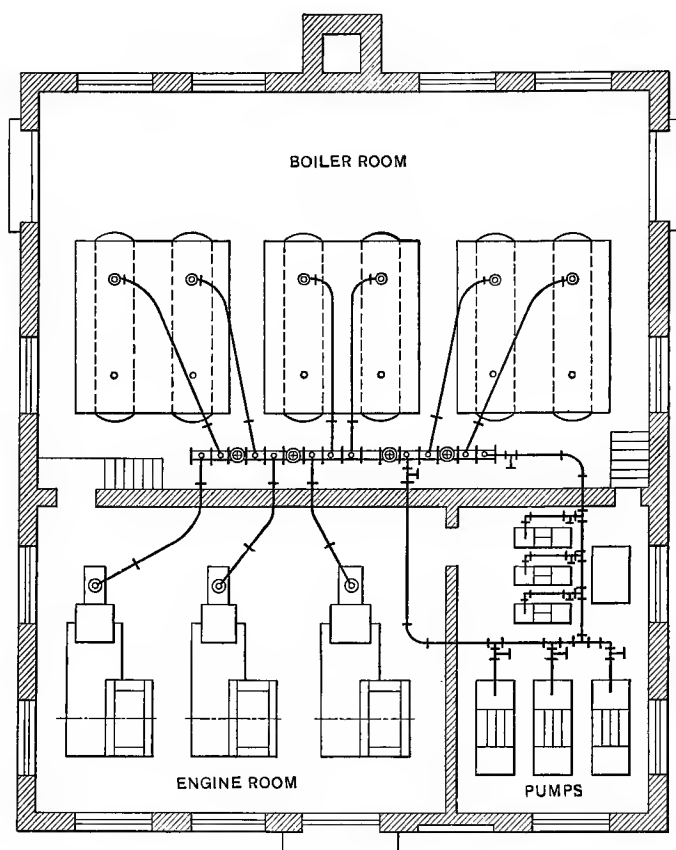


Fig. 197. Spider System of Piping.

196). This system is adapted to the same conditions as the single-loop system and is preferable in plants of large size.

349. *Spider System*.—The arrangement shown in Fig. 197 is frequently used in small plants as it shortens the main header, thus reducing the cost of construction. It also brings the prin-

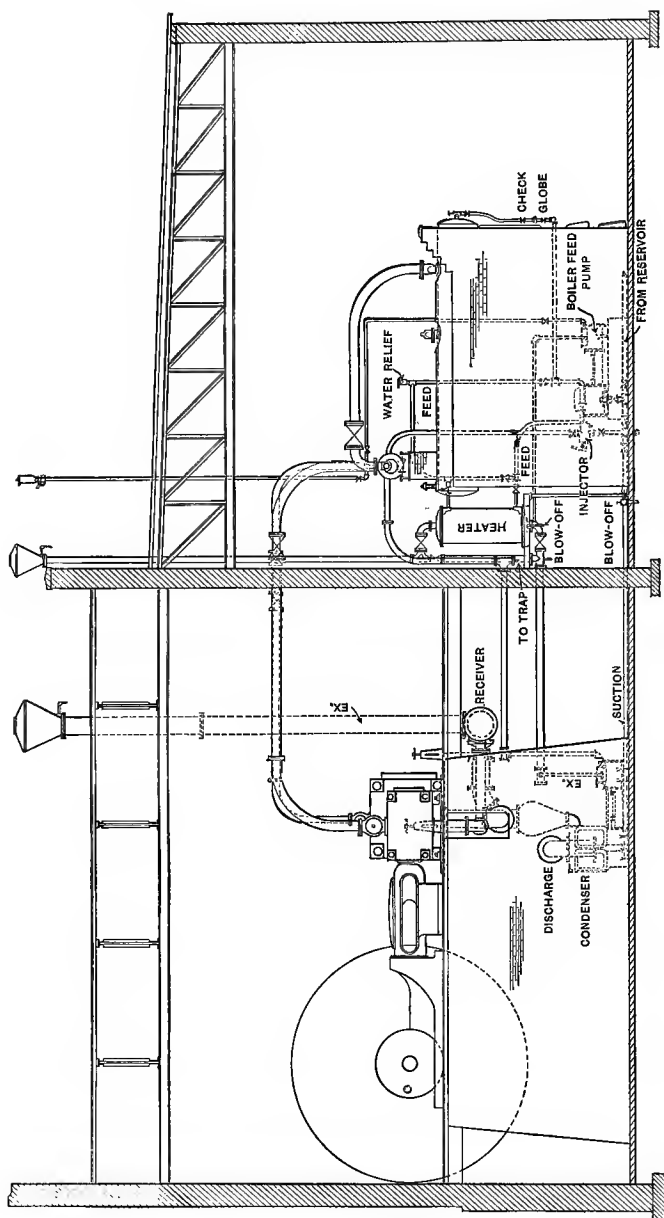


Fig. 198. Typical Arrangement for a Factory Power Plant.

cipal valves close together, which is often a matter of great convenience.

350. *Typical Systems.*—A common arrangement of equipment and piping for a factory plant, where there is ample space, is shown in Fig. 198 and is given especially for comparison with the typical office building lay-out illustrated in Fig. 199. These two cuts are shown for the purpose of emphasizing the wide variation in the arrangement of plants in different types of buildings and the necessity of making a special study of the requirements in each particular case. The essential features of a small isolated

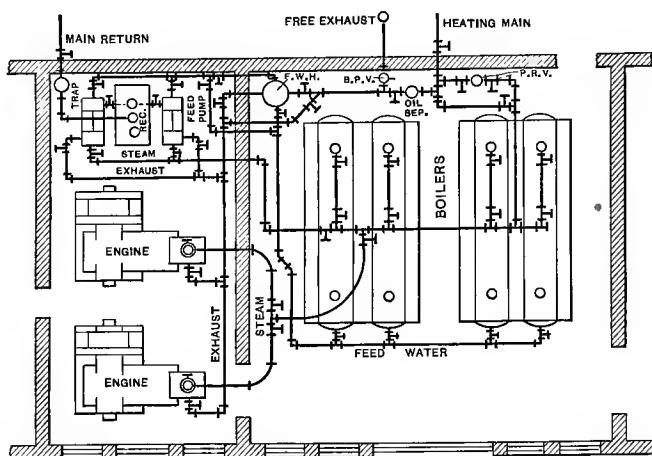


Fig. 200. Piping Diagram for a Small Power and Heating Plant.

plant arranged for heating either with live or exhaust steam are shown in Fig. 200, which illustrates the general method of connecting the piping in a plant of this kind. Live steam is fed into the system through a pressure reducing valve and the connections are so arranged that the exhaust may either be turned into the heating main or discharged outboard through a back-pressure valve. The condensation from the heating system is trapped into a vented receiver and pumped back to the boilers through a feed-water heater as indicated.

Piping Details.

The following diagrams illustrate the general principles to be followed in making the connections between the various pieces of apparatus in combined power and heating plants and will be found useful as guides in laying out new work.

351. Connections Between Boilers and Engines.—The usual methods employed for making the connections between the boiler outlets and the main header have already been shown in Figs. 179 to 182. The leads from high-pressure boilers should, in general, be provided with two valves, located in such a manner as to avoid the formation of pockets. When a non-return valve is employed, it may be made to take the place of one of the stop-

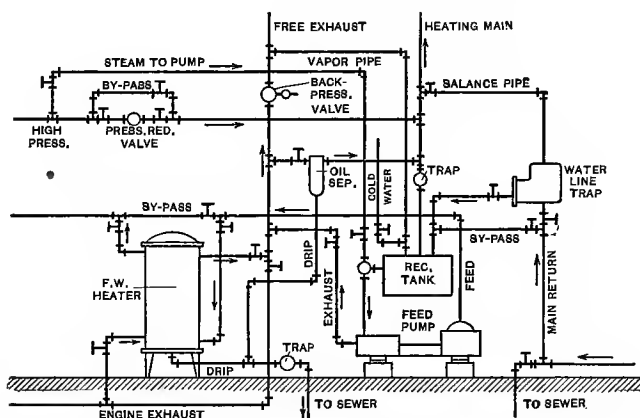


Fig. 201. Diagram of Exhaust Heating Connections.

valves. These are usually of the angle pattern and are placed directly above the boiler nozzle.

If two valves are used, as noted above, they should be so arranged that one will close against the boiler pressure and the other against the header pressure. Connections between the main header and the engines have already been described in Paragraph 343.

352. Exhaust Heating Connections.—A typical lay-out of the supply and return connections for an exhaust heating system with a gravity return to the receiving tank is shown in Fig. 201. In

a general way the arrangement is similar to that illustrated in Fig. 200 except the piping is shown in more detail. The heating main is connected for both live and exhaust steam and the base-ment returns are sealed by means of a water-line trap.

353. *Connections for Feed-Water Heaters.*—The steam connections for open heaters have already been illustrated in Fig. 154. In the case of closed heaters the general arrangement is similar, two common methods being shown in Figs. 202 and 203. In the first of these, called a “through” heater, all of the steam passes

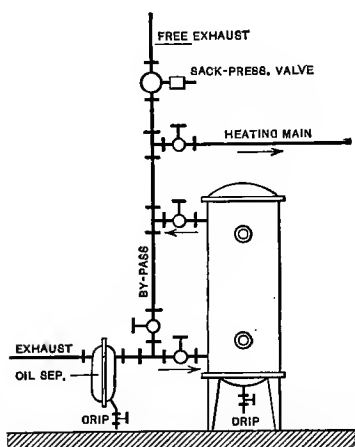


Fig. 202.

Fig. 202. Through Connection for a Feed-Water Heater.

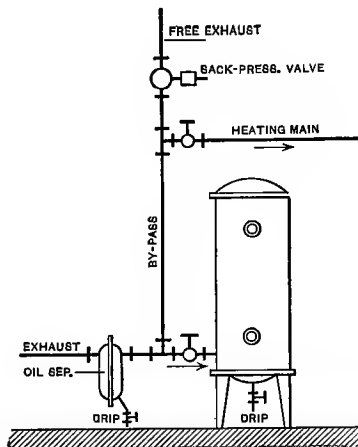


Fig. 203.

Fig. 203. Induction Connection for a Feed-Water Heater.

through the heater under ordinary conditions, although a portion may be deflected through the by-pass if desired. In the “induction” arrangement, shown in Fig. 203, the piping is simplified and the number of valves reduced by making a single steam connection with the heater. This is done in such a manner that steam will tend to be driven into it before any of the surplus passes out-board. Either method will operate satisfactorily when properly installed, and a choice between the two is usually governed by the available space or other local conditions.

354. *Connections for Jet Condensers.*—A typical arrangement of a jet condenser and a primary feed-water heater is shown in Fig. 204. The placing of the valves is such in this case that the steam may be passed directly outboard or turned through the heater into the condenser. Should it be desired to run non-condensing and turn the exhaust into the heating system it may be done by manipulating the two valves directly above the condenser. In the arrangement shown, the heater drains into the condenser without special connections. When this is not possible, and the heater must be placed at a lower level, a *return trap* is

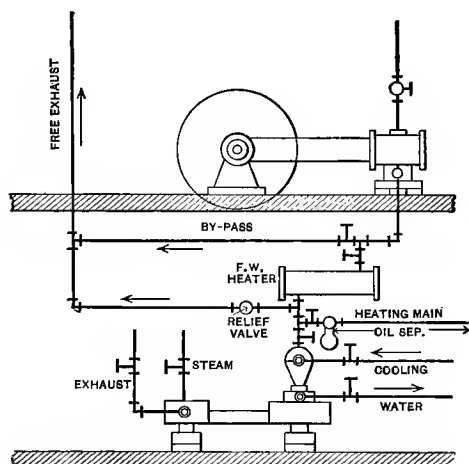


Fig. 204. Piping Diagram for a Jet Condenser and Primary Feed-Water Heater.

necessary to discharge the condensation, which is at condenser pressure, against that of the atmosphere.

355. *Connections for a Surface Condenser.*—The general method of making the steam and water connections for a condensing outfit employing a surface condenser is shown in Fig. 205. The steam connections here are similar to those for the jet condenser above described, the valve arrangement being such that the exhaust may be passed into the condenser, turned outboard, or into the heating system as desired. The condensation in this case is discharged into a hot-well and returned to the boilers

through a feed-water heater supplied with exhaust steam from the auxiliary pumps.

The heater arrangements shown in Figs. 204 and 205 are often combined in large plants where the exhaust from the auxiliaries is not sufficient for all requirements. In Fig. 205 an oil separator is placed in the exhaust line leading to the condenser. The cold feed or "make-up" water is supplied to the hot-well through a separate connection not shown in the cut.

356. Condenser Arrangement.—Two general arrangements are employed in this case, the *independent* or sub-divided system where each engine is provided with its own condensing equipment, and the *central* system in which one condenser suffices for all

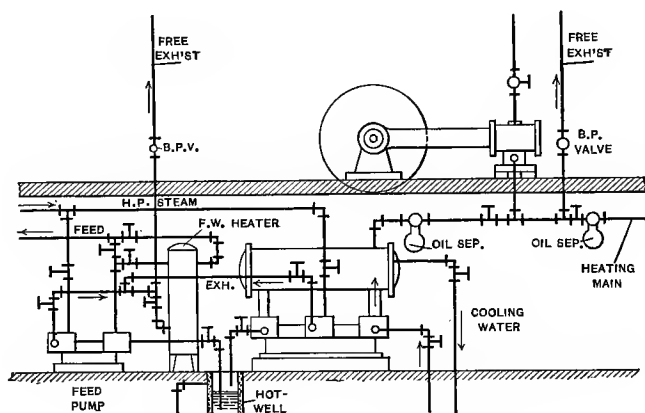


Fig. 205. Typical Piping Plan for a Surface Condenser.

of the engines. The first of these is confined principally to central stations and to plants employing large turbines. In isolated plants where the condensing equipment is used only in the non-heating season or where the requirements are more or less variable, a single condenser is usually employed as shown in Fig. 206. This arrangement is especially adapted to the conditions stated above, the connections being so made that the entire plant may be run either condensing or non-condensing, or a portion of the engines may be operated condensing while the remainder exhaust into the heating system.

357. *Location of Oil Separator.*—The location of the oil separator, with reference to the feed-water heater, will depend somewhat upon circumstances, but in general, it should be placed between the engine and heater, as shown in Figs. 202 and 203. The reason for this is to prevent the tubes from becoming coated with oil which reduces the transmission of heat through them. In some cases it is more convenient to place the separator in the heating main, as shown in Fig. 201, and allow the exhaust to pass into the heater without purification. This is sometimes done in heating plants where the quantity of "make-up" water is small and a slight lowering of the tube efficiency is not of great importance. In the arrangement illustrated in Fig. 204 the oil separator

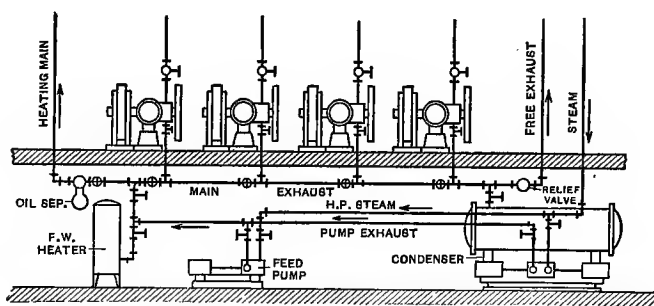


Fig. 206. Typical Piping Plan for a Central Condensing Plant.

rator might be placed in the exhaust main, above the heater, provided there was sufficient space, thus ensuring cleaner tubes, as previously noted. The lay-out shown is especially adapted to close quarters and might be made still more compact by placing the heater near the ceiling and making the exhaust connection in the side instead of the top. When the separator is located between the engine and heater, the drainage from the latter, being free from oil, may be trapped to the receiving tank. In the case of a turbine plant, no oil separator is required, except in the exhaust line from the auxiliaries.

What has been said regarding the relation between the oil separator and feed-water heater also applies to the surface condenser for the same reason.

Feed-Water Piping.

358. Method of Making Connections.—A typical arrangement of the feed-water piping for a non-condensing plant is shown in diagram in Fig. 207. Cold water from the supply main enters at the left and there branches, one line leading to the injector, which is used for emergency purposes only, and the other leading to the feed pumps. The feed-water heater is supplied with steam, both from the engines and pumps. The pumps are in duplicate, each being of sufficient capacity to do the entire work, and all equipment is by-passed as indicated.

The connections for a condensing plant are shown in Fig. 208. In this arrangement the cold water and condensate from the

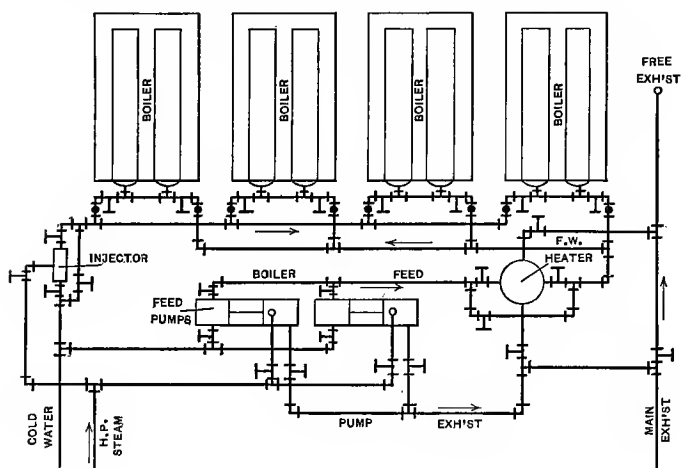


Fig. 207. Diagram of Feed System for a Non-Condensing Plant.

discharge pump are turned into a hot-well, from which the feed pumps take their supply. Both primary and secondary heaters are provided in this case, the first being placed in the main exhaust line from the engines and the second taking the pump exhaust as indicated. The injector connections are practically the same as before.

359. Feed Supply from Storage Tank.—It sometimes happens in manufacturing plants that considerable quantities of hot water are available at irregular periods, as from steam presses or other

mechanical processes. This water may often be utilized for boiler feeding by an arrangement similar to that shown in Fig. 209. A large storage tank is placed above the pump receiver and provided with an overflow and vent. The hot-water supply enters at the top, and the bottom of the tank is connected with the pump receiver by means of a ball-cock which admits water to the receiver as needed. When the hot water supply is insufficient for boiler requirements cold water is admitted to the storage tank through a ball-cock located near the bottom.

360. Boiler Feed Connections.—A diagram illustrating the different methods of placing the valves in the feed pipes to the in-

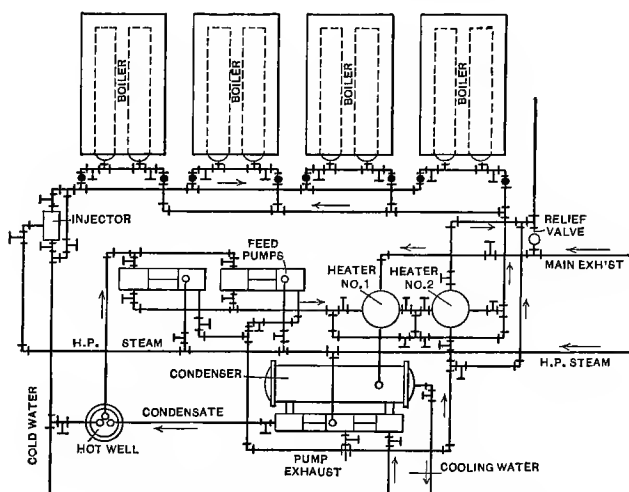


Fig. 208. Diagram of Feed System for a Condensing Plant.

dividual boilers of a battery is shown in Fig. 210, in which "A" is a stop valve, "B" a check valve, and "C" a regulating valve.

Beginning at the left, No. 1 shows the most approved arrangement where the boilers are operated continuously and the regulating valve is subjected to hard usage. In this case the stop valves are left wide open, except in case of repairs to the check or regulating valve, and are, therefore, ordinarily free from wear. In Nos. 2 and 3 one stop valve has been omitted and the regulating valve is made to take the place of a stop valve in case of re-

pairs to the check. These are the ones most commonly employed in practice. In heating work the stop and regulating valves are usually combined as in No. 4. This makes it necessary to shut off

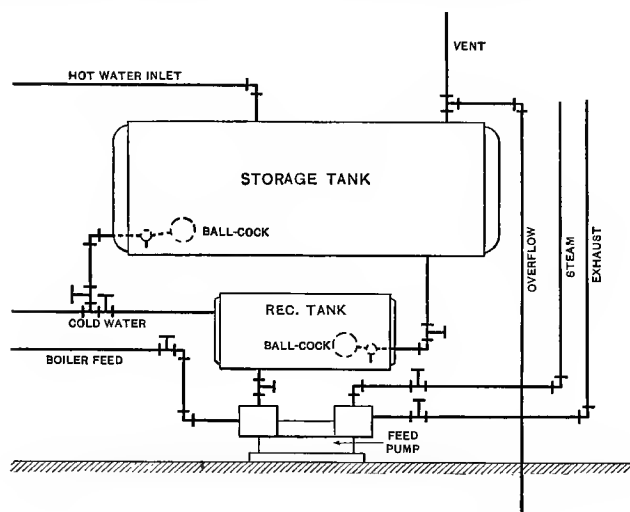


Fig. 209. Pipe Connections for a Hot-Water Storage Tank.

the pressure in the feed main in case of repairs to the check valve and should not be used in power work or in large and important heating plants. All frequently used valves in the feed pipe should

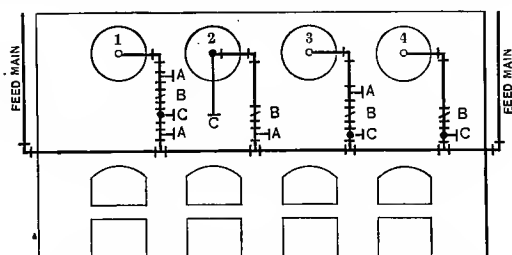


Fig. 210. Typical Valve Arrangement for Boiler Feed Pipes.

be accessible from the boiler room floor, if possible, or from a platform at the engine room level. The latter arrangement is often adopted in the case of high water-tube boilers, or extension

stems may be used as shown in connection with boiler No. 2, Fig. 210.

Size of Feed Pipe.

361. Method of Computing.—In computing the size of feed pipe for a boiler it is customary to provide for a capacity twice that required at its normal rating, and proportion the pipe for a velocity of about 100 feet per minute for sizes ranging from 1 to 2 inches. This calls for a sectional area of 0.026 square inches per boiler horse-power, which applies to the individual branches.

The main for supplying a battery of boilers may be determined from Table LXXXI., which takes into account a higher allowable velocity in pipes of larger size and the fact that all of the boilers will not require the full capacity of the pipe at the same time.

TABLE LXXXI.

EQUATION OF BOILER FEED PIPES.

Diameter of main, inches.	Number of branches of different diameters supplied by main.				
	1 inch.	1¼ inch.	1½ inch.	2 inch.	2½ inch.
2	6	4	3		
2½	9	5	4	2	
3	..	8	6	3	2
3½	..	14	10	5	3
4	..	..	14	7	4
5	..	..	20	10	5

Drainage.

362. Parts Requiring Drainage.—The drainage of condensation from the various parts of a steam system is an important detail of design and should receive careful attention. The parts requiring drainage in the average power and heating plant are the main steam header and branches to the boilers and engines, if pocketed in any way; steam and oil separators; exhaust heads; the bottoms of all risers, and all pockets in steam and exhaust mains and branches; engine cylinders and throttles; steam turbines; and closed feed-water heaters. While the condensation from the coils and radiators of a steam heating system comes under the general head of *drainage*, it is usually considered a part of the heating plant, and has already been treated in Chapter II.

363. Low-Pressure Drips.—These, as the name implies, include all drips from the exhaust system, and as they usually contain oil, are commonly discharged into the sewer. In special cases where fuel is high and water scarce, the oil is removed by separating tanks or filtration and the water returned to the boilers. As the drips from steam turbines do not contain oil they may be returned without purification. When the drainage system is connected with a city sewer the drips must be cooled to 100 degrees or less by passing through a tank; the blow-off tank from the boilers being commonly utilized for this purpose. Drips from the heating mains and risers are taken back into the general return system.

364. High-Pressure Drips.—Drips from all high-pressure piping and steam separators, being free from oil may either be trapped to the receiving tank and pumped to the boilers with the condensation from the heating system, or they may be collected in a special receiver and returned by means of a "steam loop" or "Holly System."

365. Drips from Engines, Turbines and Pumps.—Steam engines are dripped from each end of the cylinder, and from the steam pipe just above the throttle, unless a vertical separator is used close to the cylinder. These drips are only opened for a short time when starting up and are commonly connected into the exhaust pipe. The drainage of steam turbines varies with the type and construction, and should be designed with special reference to directions obtained from the makers. Steam pumps which are started by hand require the same treatment in the matter of drainage as an engine. Elevator pumps which work intermittently and are started automatically by changes in the pressure tank, should have the throttle drip connected with a trap in order to keep it clear continuously.

366. Drip Connections for Steam Mains.—Steam mains and branches in which the flow is always in the same direction should have a pitch of 1 inch for each 20 to 40 feet of length. Headers in which the flow may be in either direction should be placed level and dripped at each end. Some engineers use a single drip at the end of the line regardless of its length, while others make a connection every 125 or 150 feet with a drain pipe which is carried below the steam main back to the boiler room.

The best method to employ in any given case will depend upon local conditions, such as the probable amount of condensation, the method of taking off leads to engines and pumps, and other details of a similar nature. In any case the object sought is the removal of water from all pockets and low points without surging or water hammer. When superheated steam is used the mains and headers should be drained at frequent intervals as the presence of water in the bottom of the pipe tends to produce an unequal temperature which is liable to distort it and cause leakage at the joints.

The ends of large mains and the bottoms of risers are most effectively drained by providing a pocket the full size of the pipe, three or four diameters in depth, and connecting a drip pipe with the lower end. Where a pipe is reduced in size an eccentric fitting should be used to keep the bottom of the line level. If this is not possible, the larger section of pipe should be dripped at the point of reduction.

367. Sizes of Drip Lines.—The size of drip will depend somewhat upon the length of the main, but for the conditions found in plants of medium size the following may be used:

TABLE LXXXII.

Diameter of main, inches.	Diameter of drip, inches.
2	$\frac{3}{4}$
4	1
8	$1\frac{1}{4}$
12	$1\frac{1}{2}$
16	2

368. Traps.—The different types of steam traps employed in power and heating work have been described in detail in a previous volume. Non-return traps, so-called, are used in all cases where condensation is to be discharged from a higher to a lower pressure. Return traps are required where the conditions are reversed, as in returning the condensation from a low-pressure heating system into a high-pressure boiler, and the draining of vacuum feed-water heaters against atmospheric pressure.

Outboard exhaust risers are often drained through siphon loops of sufficient depth to balance the difference in pressure.

369. *Blow-Off Connections.*—Each boiler in a battery should be so piped that it may be blown off independently of the others. The size of connection is usually fixed by the makers, which also determines the size of the main pipe leading to the tank, as only a single boiler is blown off at once. Each connection should be provided with a valve especially designed for this purpose, supplemented by a gate valve for use in case of leakage or accident to the blow-off valve.

CHAPTER X.

COMBINED POWER AND HEATING.

Previous chapters have taken up the heating, ventilating and power requirements in different types of buildings independently, each being treated without special reference to the others.

370. Combined Power and Heating.—In the present chapter the entire plant will be considered as a whole and all requirements involving the ultimate use of steam will be studied in connection with their relation to one another. The practical application of this will occur in the design of new plants for power and heating, the addition of power equipment to an existing heating plant, and the utilization of exhaust steam for heating from a power plant already in use.

After discussing certain conditions which are common to a greater or lesser degree in all isolated plants, a number of the more usual types of buildings will be considered in detail, showing what changes in piping and apparatus are necessary for converting a simple heating plant into a combined heating and power plant, and vice versa. Later, the cost of such changes will be studied to some extent, both from the point of installation and operation. The fact that conditions vary so widely in different types of buildings, and even in buildings of the same type, makes it impossible to give much general data which can be safely used broadcast. The method followed has been to assume specific conditions and work the problem out on that particular basis. While the assumptions for different buildings are sufficiently accurate to give the results some general value as applying to that particular class, the chief use of these problems is to illustrate the general methods to be followed. By substituting actual values from the conditions involved in any particular case results may be worked out with a good degree of accuracy.

In general, it is usually economy to install lighting plants in large buildings and employ the exhaust steam for heating, or to utilize the exhaust for this purpose from power plants already in

use. While this may be true in a general way the *amount* of saving will depend upon a considerable number of factors, such as relation of steam required for heating to the supply of exhaust at different parts of the day, length of the heating season, type of engines used, method of heating, cost of fuel, water, labor, etc. The total exhaust from a plant in twenty-four hours may be equal to the heating requirements during the same period, but if it is not properly distributed throughout the day a large amount may be thrown away at certain times which must be made up later by using live steam when the power output is at a minimum. Hence it is seen that a special study must be made of each individual case in order to determine the best possible arrangement and the final results as regards average economy of operation throughout the year.

Available Exhaust.

371. Heat in Exhaust Steam.—The heat units carried away in the exhaust are equal to the total heat in the steam at admission, less that transformed into work and lost in cylinder condensation. This varies somewhat with the initial pressure, ratio of expansion and type of engine, but in the average case it is customary to assume that 80 to 90 per cent of the heat in the steam supplied to the engine is available in the exhaust for heating purposes. Another method which gives results similar to the above, is to assume an exhaust pressure of 1 to 5 pounds gauge, and estimate the weight of steam discharged from the engine as 80 to 85 per cent of that supplied. Knowing the latent heat of steam at exhaust pressure, (see Table LXII.), the total heat value is easily determined in any given case.

372. Surplus Exhaust.—Unless there is some way of storing the heat from exhaust steam, any surplus which may be produced at different periods of the day must be thrown away, as previously stated and is therefore of no value. For this reason it is necessary to make a careful comparison of the exhaust produced and the heat required for each hour of the day, in order to determine how much of the total exhaust is *available* and how much *surplus* or waste. Therefore, in all problems relating to combined power and heating, *available* exhaust and not the *total* is the important factor to be considered.

Power Curves.

373. Method of Plotting.—The first step in making a comparison of this kind is to determine the average indicated horse power required for all power purposes for each hour of the day and plot a curve which shall represent the pounds of exhaust steam produced, expressed in the equivalent of live steam at the heating pressure to be used, which commonly runs from 1 to 5 pounds gauge.

374. Power Requirements in Old Plants.—If the power requirements are for a plant already in use it is a simple matter to determine them quite accurately for each hour in the day if engine room logs have been carefully kept for one or more preceding seasons. If the problem relates to a building having its own heating system, but buying electricity for power and lighting purposes from a central plant, the necessary data can be obtained from the meter records, provided they are read at hourly intervals, or have been automatically recorded. With this data at hand and knowing the type of engine it is proposed to install, the average quantity of exhaust steam can be quite accurately estimated.

375. Power Requirements in New Plants.—In the case of a new plant in process of design, careful estimates must be made according to the proposed conditions of operation. For example, if the plant is for an office building, requiring power for lighting, and elevator service only, an estimate should be made of the probable number of lights in use for each hour in the day, and the indicated horse power of generator engine required for supplying them. The approximate power for operating the elevators should be taken up in a similar manner. The period of lighting will vary with the season of the year, and to some extent with the condition of the weather as to cloudiness. The hours for general lighting are easily obtained, together with the number of lamps which are burned all day. The proportion of cloudy weather to be provided for may be approximated from the weather reports of previous years.

Power for elevator service can be estimated in buildings of this kind by assuming maximum requirements for an hour or so from 8 until 9 in the morning, again from 12 until 2, and again from 5 until 6, or such other hours as local custom may indicate.

During the remainder of the day the elevators may be assumed to run at one-half to one-fourth capacity, according to the character of the business conducted in the building. In the case of a large store the power requirements would be more uniform.

In many departments lights are on continuously and elevators operated at approximately full capacity throughout the day, or at least the service called for is nearly uniform. If ventilating fans and pneumatic cash systems are in use, the power required for these is practically constant. Attention is called to these particular buildings simply for purposes of illustration, and a careful study must necessarily be made of each plant under consideration and every item of power determined so far as possible both as to time and amount.

376. Power Requirements for the "Average Day."—While hourly averages are necessary in most buildings on account of the wide fluctuation of power requirements, daily reports are not so important, as the change from day to day is comparatively slight. For example, a good arrangement is to plot a curve for an *average* day representing the six working days of each week. Another curve may be plotted for Sunday if conditions are such as to require it for a proper study of the problem.

A simpler method, and one which is sufficiently accurate for many cases, is to divide the heating season into months and compute the power and heating requirements for an average day in each. In a climate similar to that of New England, the heating season may be considered as beginning the middle of October and lasting until the middle of May, in average years. This divided into seven equal periods forms a good basis upon which to make computations of this kind unless very accurate results are required.

Steam Curves.

377. Method of Plotting.—After plotting the power curve, in terms of the exhaust, the various steam requirements, other than for power, should be determined in a similar manner for each hour in the day and plotted on the same diagram. This shows at once the quantity of exhaust steam being furnished by the engines and that required for all heating purposes at any period of the day.

378. Steam Requirements for Heating.—The steam required for heating is not so easily obtained as that for power owing to the fluctuation of the outside temperature, variation in the strength and direction of winds, and the proportion of sunshine. The radiation in a building is designed for the coldest weather of the season, commonly taken as zero in the temperate zone. As the outside temperature may vary anywhere between this point and 70 degrees above during the heating season, it is evident that the steam requirements cannot be based upon supplying the full amount of surface throughout the entire season.

If the radiation could be equipped with a perfect system of automatic control so that an inside temperature of 70 degrees would be maintained at all times, the quantity of steam used would be in direct proportion to the difference between the inside and outside temperatures. Then by consulting weather charts for previous years approximate averages could be obtained for a series of days, weeks or months, as desired, and the probable steam requirements estimated with a considerable degree of accuracy. On the other hand, if all of the radiators were kept turned on throughout the heating season and the temperature of the rooms maintained at 70 degrees by opening and closing the windows, the system would be kept working at practically its maximum capacity regardless of outside temperatures, and the steam consumption would be so nearly constant that it could be easily determined.

As a matter of fact, neither of these conditions actually prevails in practice and the only way to estimate the quantity of steam required in any given case is to assume perfect regulation of temperature in the building and base all computations upon average outside temperatures taken from the weather charts of several preceding years. Results obtained in this way may then be corrected for the system of regulation employed by means of factors taken from the curves in Fig. 211. The method of determining these factors and their application to actual problems is taken up later in some detail.

379. Steam Requirements for the "Average Day."—In making up an "average day" for each week or month of the heating season, the average steam requirements for the same hour in each day of the given period should be computed. While this may

seem at first to involve considerable labor, it is not difficult with the records at hand, and will be well worth the time expended by placing the later estimates upon a logical basis instead of trusting to rough approximations.

380. *Correcting for Outside Temperature.*—Having made out a list of outside temperatures, from the weather charts, the next

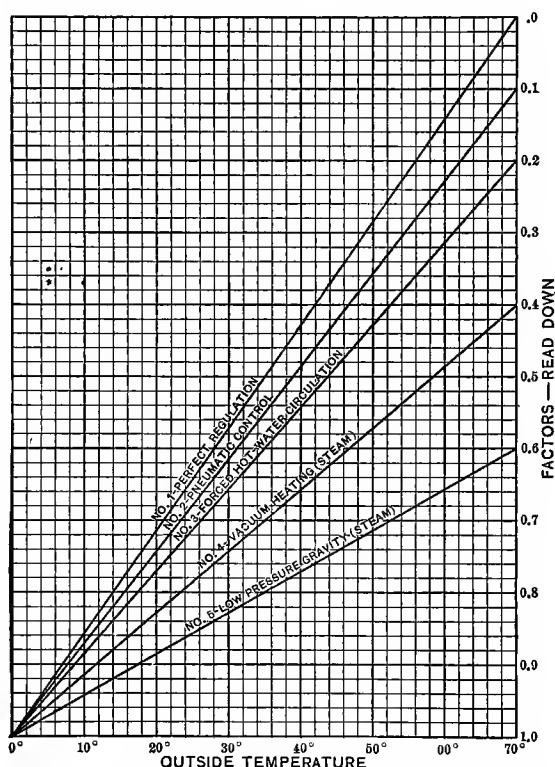


Fig. 211. Factors for Different Methods of Temperature Regulation.

step is to determine the quantity of heat required for warming the building under these varying conditions.

In the absence of strong winds the heat loss from a building is nearly proportional to the difference between the inside and outside temperatures. For example, if the heat loss from a given room is 10,000 B. T. U. per hour when it is zero outside and 70 de-

degrees above inside, the loss with an outside temperature of 35 degrees will be $\frac{35}{70} \times 10,000 = 5,000$ B. T. U., or only one half as much, and for 50 degrees above, $\frac{20}{70} \times 10,000 = 2,857$ B. T. U., and so on.

Referring to the chart, Curve No. 1 gives a simple means of finding the heat loss for any outside temperature when that at zero is known, and the inside temperature remains constant at 70 degrees. The use of the curve may be illustrated as follows: A building contains 5,000 square feet of direct radiation, which are required to warm it in zero weather, it being assumed that each square foot of surface gives off 250 B. T. U. per hour when the rooms are at 70 degrees. On this basis how many pounds of steam at 5 pounds pressure will be required per hour when the outside temperature is at the following points: Zero, +10 deg., +20 deg., +30 deg., +40 deg., +60 deg. and +70 deg.?

Referring to Curve No. 1, Fig. 211, we find the following factors or multipliers:

0 deg.	— 1.00
+ 10 deg.	— 0.86
+ 20 deg.	— 0.72
+ 30 deg.	— 0.57
+ 40 deg.	— 0.43
+ 50 deg.	— 0.29
+ 60 deg.	— 0.15
+ 70 deg.	— 0.00

The latent heat of steam at 5 pounds pressure is 960 B. T. U., hence, the weight required at zero temperature outside is

$$\frac{5,000 \times 250}{960} = 1,302$$

pounds, or 1,300 pounds in round numbers. Correcting this for other outside temperatures by the above factors, we have the following results:

Outside Temperature.	Pounds of Steam Per Hour.
0 deg.	$1,300 \times 1.00 = 1,300$
+ 10 deg.	$1,300 \times 0.86 = 1,118$
+ 20 deg.	$1,300 \times 0.72 = 936$
+ 30 deg.	$1,300 \times 0.57 = 741$
+ 40 deg.	$1,300 \times 0.43 = 559$
+ 50 deg.	$1,300 \times 0.29 = 377$
+ 60 deg.	$1,300 \times 0.15 = 195$
+ 70 deg.	$1,300 \times 0.00 = 0$

Corrections for Actual Conditions.

381. Effect of Temperature Regulation.—The steam consumption for heating under actual conditions will vary from the ideal described above according to the system used and the method of regulation employed. With the best systems of pneumatic control an average inside temperature of 70 degrees may be approximately maintained, so that only the radiating surface in the mains and risers will be in service when the outside temperature reaches the same point. This method of control, which may be applied either to steam or hot-water heating, is the most efficient in use and approaches to a certain extent the ideal conditions given by curve No. 1.

Next in point of efficiency, as regards regulation, is the system of forced hot-water circulation. This is under the direct control of the engineer and the water may be forced through the radiators at a temperature depending upon that of the outside air. It is necessary, of course, to carry a water temperature sufficiently high to warm the coldest section of the building, hence, there will be more or less heat wasted through open windows in the warmer rooms, as is always the case when hand regulation by the occupants is depended upon.

Next in order is the return-line vacuum system equipped with graduated hand-valves. The efficiency of regulation in this case depends so largely upon the care exercised by the individual occupants that a considerable margin must be allowed for wasted heat.

In the remaining system, still used to some extent, steam is kept on the building practically all of the time throughout the heating season, and the tenants regulate the temperature of their rooms either by manipulating the radiator valves or by opening and closing the windows, usually the latter. This arrangement is necessarily wasteful in the use of steam and the requirements are hard to estimate.

382. Factors for Temperature Regulation.—The determining of suitable factors for offsetting the steam wasted with different systems of regulation requires a considerable amount of practical observation and results will be found to vary somewhat in the experience of different engineers.

In the absence of more exact information, curves 2 to 5 in Fig. 211, may be used for the systems of control there noted.

These factors are for average conditions and will vary considerably, depending upon the care and skill of the operating engineer and the extent to which the occupants make use of the radiator valves, graduated or otherwise.

They are based upon the assumption that when the outside temperature reaches $+70$ degrees, 10 per cent of the heating capacity of the plant will be in use with automatic regulation; 20 per cent with forced hot water; 40 per cent with vacuum systems, and 60 per cent with low-pressure gravity. In other words, with automatic regulation, all radiation except the mains and risers will be shut off at 70 degrees; with forced hot water the plant will be working at 20 per cent of its capacity; with vacuum systems at 40 per cent, and with gravity steam, 60 per cent of the radiators will still be turned on.

Of course, under actual conditions the engineer will shut off heat from the building when the outside temperature reaches 70 degrees, whatever the system in use, but the same relations hold for all temperatures between zero and 70 degrees.

Applying these corrections to the preceding example, for an outside temperature of $+40$ degrees, it gives the following steam requirement in pounds per hour for different methods of heating:

Pneumatic control.....	1,300	$\times 0.48 =$	624
Forced hot water.....	1,300	$\times 0.54 =$	702
Vacuum system.....	1,300	$\times 0.66 =$	858
Low-pressure steam (gravity)....	1,300	$\times 0.77 =$	1,001

Practical Example in the Use of Power and Steam Curves.

Having shown the general method of collecting and arranging the data for making a comparison between the available exhaust and the steam required for heating, let us now work out a practical example to show its application and also plot the two curves.

Take the case of an office building requiring power for lighting, elevator service, auxiliary pumps and fan motors, and steam for heating, ventilation, and hot-water supply.

383. General Assumptions.—Let the maximum power required

for lighting be 100 Kw.; for elevator service, 150 I. H. P.; for fan motor, 6 I. H. P., and for the various auxiliary pumps, 10 I. H. P.; all referred to the main engine.

Let the power required for lighting on an average day for the assumed week be as follows, for the different hours which the building is to be occupied, either by tenants, attendants, or cleaners, assuming that each kilowatt at the generator requires 1.75 I. H. P. at the engine.

Hour of day.	Kw. for lighting.	I.H.P. of engine.
5 a. m. to 7 a. m.	5	9
7 a. m. to 8 a. m.	25	44
8 a. m. to 4 p. m.	35	62
4 p. m. to 6 p. m.	100	175
6 p. m. to 9 p. m.	20	35

From 5 to 7 a. m. just enough lights are on for the use of attendants, etc.; from 7 to 8, corridors and some additional rooms are lighted; from 8 a. m. to 4 p. m. the regular day lighting is to be provided for, which includes basement, corridors, store lighting, etc.; from 4 to 6, the entire building is lighted, and from 6 to 9 a portion of the lights are on for purposes of cleaning, repairing, etc.

For elevator service the following may be assumed:

Hour of day.	I. H. P. of engine.
7 a. m. to 8 a. m.	50
8 a. m. to 9 a. m.	150
9 a. m. to 12 noon.	75
12 noon to 2 p. m.	150
2 p. m. to 4 p. m.	50
4 p. m. to 5 p. m.	150
5 p. m. to 6 p. m.	75
6 p. m. to 7 p. m.	25
7 p. m. to 9 p. m.	5

The ventilating fan is run continuously from 8 a. m. till 5 p. m. for first floor stores, and produces a steady load of 6 I. H. P. at the engine.

The building is heated by forced hot water and the auxiliary pumps, including those for hot-water circulation, run from 6 a. m. till 9 p. m., requiring the equivalent of 10 I. H. P. at the main engine.

The engines used for driving the generators are of the simple,

high-speed, non-condensing type, with the units so divided that they run with fair economy throughout the day, and an average water-rate of 30 pounds of steam per I. H. P. per hour may be assumed. It is further estimated that the heating equivalent of the exhaust is equal to 85 per cent of the weight of steam supplied to the engines, at 5 pounds gauge pressure, as previously described for conditions of this kind. That is, in the present case, the equivalent of $30 \times 0.85 = 25.5$ or 25 pounds of steam in round numbers at 5 pounds pressure will be available for heating purposes in the exhaust, per hour, for each I. H. P. of the main engines.

384. Power Curve.—Working out the above computations separately for each hour of the day for the total I. H. P. in use for all purposes during that time we have Table LXXXIII., in which the first column under "I. H. P. for different purposes" gives the power required for lighting; the second, for elevator service; the third, for auxiliary and circulating pumps; and the fourth, for ventilating fans.

The last column in the table gives the pounds of exhaust steam available for heating purposes during each hour of the day, and is found by multiplying the total I. H. P. by 25.

TABLE LXXXIII.

Hour of the day.	I.H.P. for different purposes.	Total I.H.P.	Pounds of exhaust steam per hour.
5 to 6	9	9	225
6 " 7	9 + 10	19	475
7 " 8	44 + 50 + 10	104	2,600
8 " 9	62 + 150 + 10 + 6	228	5,700
9 " 10	62 + 75 + 10 + 6	153	3,825
10 " 11	62 + 75 + 10 + 6	153	3,825
11 " 12	62 + 75 + 10 + 6	153	3,825
12 " 1	62 + 150 + 10 + 6	228	5,700
1 " 2	62 + 150 + 10 + 6	228	5,700
2 " 3	62 + 50 + 10 + 6	128	3,200
3 " 4	62 + 50 + 10 + 6	128	3,200
4 " 5	175 + 150 + 10 + 6	341	8,525
5 " 6	175 + 75 + 10	260	6,500
6 " 7	35 + 25 + 10	70	1,750
7 " 8	35 + 5 + 10	50	1,250
8 " 9	35 + 5 + 10	50	1,250

Curve No. 1 shown in Fig. 212 is plotted from these figures.

385. Steam Curve.—The outside temperatures for an "average day" for the particular week chosen for illustration are made up from the weather reports of preceding years, and are as shown in

the second column of Table LXXXIV. Column 3 gives the corresponding multipliers taken from curve No. 3, Fig. 211, for forced hot-water heating. Suppose the total B. T. U. required for warming the building in zero weather is found by computation to be 1,920,000 B. T. U. per hour. Heat is turned on at 6 o'clock in the morning, and it is assumed that when the temperature is less than $+20$ degrees outside it will take 2 hours to warm the building up to 70 degrees; hence, the heating system

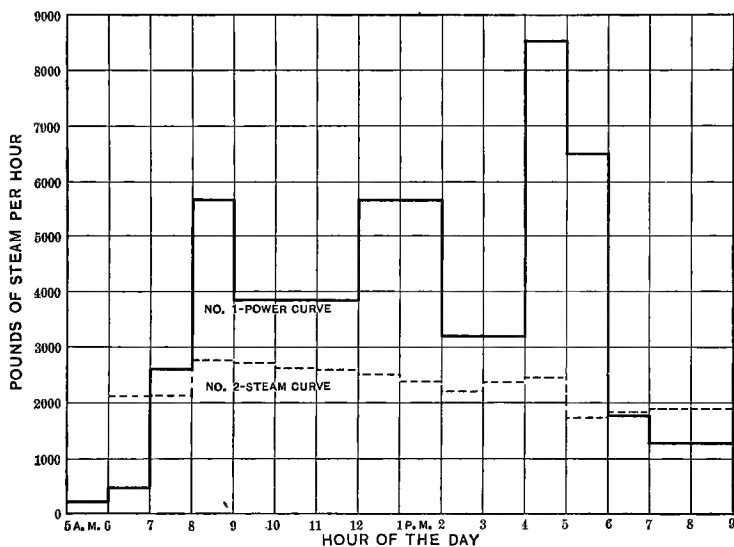


Fig. 212. Method of Plotting Power and Steam Curves.

will be working at its full capacity from 6 until 8 a. m., regardless of the outside temperature under the conditions of the present problem.

Heat is shut off from the building at 9 p. m., at the same time the engines are shut down—that is, the circulating pumps are stopped at that time. Steam is required for warming the air for ventilation from 8 a. m. until 5 p. m. The heat required for this purpose per hour is assumed to be 960,000 B. T. U. in zero weather, and varies with the outside temperature.

The hot-water storage tank is of large size and steam is on the coil from 6 a. m. until 9 p. m. It is further assumed that 96,000

B. T. U. per hour are used continuously for water heating throughout the day. This covers all uses of hot water both for lavatory and cleaning purposes.

Although the quantity used will vary at different hours of the day the large size of the storage tank serves as a reservoir and distributes the steam required for this purpose over the entire period. The quantities in column 4 are obtained by the equation $\frac{1,920,000}{960} \times F$ in which F is the multiplier from column 3 corresponding with the outside temperature for that hour. The quantities in column 5 are obtained by the equation $\frac{960,000}{960} \times F$, in which F is taken the same as before for simplicity. The quantities in column 6 are equal to $\frac{96,000}{960}$ and are uniform throughout the day.

Column 7 gives the total steam requirements per hour for all heating purposes, being made up of the quantities in columns 4, 5 and 6.

TABLE LXXXIV.

1.	2.	3.	4.	5.	6.	7.
Hour of the day.	Outside temperature, degrees.	Multiplier.	Pounds steam per hour for heating.	Pounds steam per hour for ventilation.	Pounds steam per hour for hot-water service.	Total pounds steam required per hour.
5 to 6	+ 8	...	2,000	...	100	2,100
6 " 7	+ 8	1.00	2,000	...	100	2,100
7 " 8	+10	1.00	2,000	...	100	2,100
8 " 9	+10	0.89	1,780	890	100	2,770
9 " 10	+11	0.88	1,760	880	100	2,740
10 " 11	+12	0.86	1,720	860	100	2,680
11 " 12	+14	0.84	1,680	840	100	2,620
12 " 1	+18	0.80	1,600	800	100	2,500
1 " 2	+22	0.75	1,500	750	100	2,350
2 " 3	+26	0.70	1,400	700	100	2,200
3 " 4	+22	0.75	1,500	750	100	2,350
4 " 5	+20	0.77	1,540	770	100	2,410
5 " 6	+16	0.82	1,640	...	100	1,740
6 " 7	+14	0.84	1,680	...	100	1,780
7 " 8	+13	0.85	1,700	...	100	1,800
8 " 9	+13	0.85	1,700	...	100	1,800

Curve No. 2 in Fig. 212 is plotted from column 7, and is shown as dotted in order to distinguish it from the exhaust or power curve.

386. *Conditions Shown by the Curves.*—When the steam curve (No. 2), is at the top, the vertical distance between the two curves at any point shows the live steam required at that time. For example, from 6 to 7 a. m. $2,100 - 475 = 1,675$ pounds of live steam are required for heating purposes. When the power curve (No. 1), is at the top, the weight of exhaust steam represented by the vertical distance between the two, plus that lost in work and cylinder condensation, represents the steam wasted at that point. For example, from 8 to 9 a. m. the required power is 228 I. H. P. and the cylinder losses $228 \times (30 - 25) = 1,140$ pounds, making the total waste $(5,700 - 2,770) + 1,140 = 4,070$ pounds.

Strictly speaking, the term “waste” is incorrect, as this quantity includes the heat transformed into work in the engine; but as this is only a small proportion of the total, it simplifies matters to consider the entire cylinder loss as waste steam. Furthermore, as both the outboard exhaust and cylinder condensation are charged to the power account in a combined plant, there is no object in attempting to separate the different items in making the computations.

Exhaust steam used for heating feed water is included in *exhaust wasted*, because when the steam plant is operated for heating alone all condensation is returned to the boilers and there is practically no cold feed to be warmed. Therefore, when the exhaust from an added power plant exceeds the heating requirements, cold feed water must be supplied to take the place of the exhaust thrown outboard, and the surplus exhaust steam used in heating this is wasted so far as the original independent heating plant is concerned. In other words, the steam used for feed water heating is not available for warming the building, and must be charged to the power account.

For practical purposes the power and steam curves should be drawn to a much larger vertical scale, but those shown serve to illustrate the method of plotting them. Accurate quantities are best taken from Tables LXXXIII. and LXXXIV., while the curves show at a glance the general relation between supply and demand during the entire day.

Estimate of Cost.

Let us now take up the plant and determine the cost of power for a heating season of 31 weeks from October 7 to May 11.

387. *Fuel for Power During Heating Season.*—The building must be supplied with steam for heating, ventilation and hot-water service under any condition, so the additional cost for power, so far as fuel alone is concerned, is represented by the heat lost in the exhaust steam which is in excess of the heating requirements and therefore wasted.

Let power and steam curves be plotted for an "average day" in each of the 31 weeks, as already described, and suppose the pounds of exhaust wasted each day, including cylinder losses, to be as given in columns 2 and 4 of Table LXXXV. (These curves are not shown, being similar to those in Fig. 212, but are referred to in the following example):

TABLE LXXXV.

Week ending	Pounds of exhaust steam wasted per day.	Week ending	Pounds of exhaust steam wasted per day.
Oct. 14.....	38,300	Feb. 3.....	25,200
" 21.....	39,400	" 10.....	22,800
" 28.....	33,800	" 17.....	22,300
Nov. 4.....	29,100	" 24.....	22,300
" 11.....	27,700	Mar. 2.....	24,300
" 18.....	29,600	" 9.....	26,700
" 25.....	27,200	" 16.....	24,300
Dec. 2.....	27,200	" 23.....	26,700
" 9.....	28,100	" 30.....	27,500
" 16.....	26,500	Apr. 6.....	26,200
" 23.....	27,700	" 13.....	30,800
" 30.....	28,100	" 20.....	31,700
Jan. 6.....	24,300	" 27.....	32,100
" 13.....	25,200	May 4.....	34,500
" 20.....	22,800	" 11.....	38,900
" 27.....	20,400		
Total.....			871,700

The total given at the end of the last column represents the pounds of exhaust steam wasted for one day in each of the 31 weeks, from which it is evident that the total loss for six days in each week, or for the entire heating season, is $871,700 \times 6 = 5,230,200$ pounds. No account is taken of Sundays because the power plant will not be running, and there will be no exhaust wasted whether the building is heated or not.

In estimating the heat lost in "exhaust wasted," only the latent heat is considered, because that required to raise the temperature

of the cold feed to approximately the boiling point is provided by exhaust steam which is also included in this quantity, as noted in Paragraph 386. Under these conditions the total heat chargeable to power during the heating season is $5,230,200 \times 960 = 5,020,992,000$ B. T. U.

Assuming the heat value of the coal to be 13,300 B. T. U. per pound, and a boiler efficiency of 60 per cent, $5,020,992,000 \div (13,300 \times 0.6) = 629,197$ pounds of coal will be required, or $629,197 \div 2,000 = 314$ tons.

388. Fuel Cost of Power for Heating Season.—Suppose the curves referred to show that the power required for all purposes during the heating season runs from 2,000 to 2,400 horse-power hours per day, according to the time of year, or an average of 2,200. This for the entire heating season amounts to $2,200 \times 6 \times 31 = 409,200$ horse-power hours, or $409,200 \div 1.75 = 233,828$ kilowatt hours, at a fuel expenditure of $629,197 \div 233,828 = 2.7$ pounds per kilowatt hour. The cost of coal per kilowatt hour at different prices is given below:

COST OF COAL FOR POWER DURING HEATING SEASON.

Price per ton.	Cost per kilowatt hour.
\$2.00	\$0.0027
3.00	0.0040
4.00	0.0054
5.00	0.0067
6.00	0.0081

This, of course, refers to additional fuel alone and does not take into account the cost of plant, labor, etc.

389. Fuel Cost of Power for Non-Heating Season.—For the remainder of the year suppose the power requirements are found to average 2,000 horse-power hours per day, which calls for $2,000 \times 30 = 60,000$ pounds of steam. If the feed water is heated to a temperature of 204 degrees by the exhaust and the engines are supplied with steam at 90 pounds gauge pressure, the heat required per pound of steam generated will be $(331 - 204) + 885 = 1,012$ B. T. U., or $1,012 \times 60,000 = 60,720,000$ B. T. U. per day, calling for $60,720,000 \div (13,300 \times 0.6) = 7,609$ pounds of coal. This amounts to $7,609 \div 2,000 = 3.8$ pounds per horse-power hour, or $3.8 \times 1.75 = 6.65$ pounds per kilowatt hour, the cost for which, at different prices, is as follows:

COST OF COAL FOR POWER DURING SUMMER SEASON.

Price per ton.	Cost per kilowatt hour.
\$2.00	\$0.0066
3.00	0.0097
4.00	0.0133
5.00	0.0166
6.00	0.0200

390. *Fuel Cost of Power per Year.*—The average cost of fuel for power during the *year*, which is, of course, the length of time to be considered in making an estimate of this kind, is found as follows: Take for example, coal at \$3 per ton. The cost of power during the heating season of 31 weeks is \$0.004 per kilowatt hour and during the non-heating season of 21 weeks \$0.0097 per kilowatt hour, the average of which is

$$\frac{(31 \times 0.004) + (21 \times 0.0097)}{52} = \$0.0063 \text{ per kilowatt hour.}$$

In this estimate no account has been taken of the exhaust utilized for hot-water service during the summer months. This would actually amount to about 1,000 pounds per day, at the most, which is so small in comparison with the total of 60,000 pounds supplied to the engines that it has been omitted in the present case. In hotels, etc., where this is a larger item, it should be taken into account in the summer estimates as well as in the winter.

391. *Total Cost of Power per Year.*—In estimating the actual cost of power in this case we must take into account interest, depreciation and taxes on the first cost of the power plant; also repairs, supplies and additional labor involved in its operation. If water is purchased at city rates, which would naturally be the case with an office building, this must also be included. Let an approximate estimate of the cost now be made to show the general method of considering this part of the problem.

For heating, ventilating and hot-water service the total steam requirements in zero weather amount to 3,100 pounds per hour, which calls for approximately $3,100 \div 30 = 103$ boiler H. P. In case of the power plant the maximum called for is

$$\frac{(175 + 150 + 6 + 10) \times 30}{30} = 341 \text{ H. P.}$$

As the available exhaust always exceeds the heating require-

ments under maximum operating conditions, the total boiler capacity is fixed by the quantity of steam used by the engines. Hence, the *additional* boiler power called for is $341 - 103 = 238$ H. P. The cost of plant may then be taken approximately as follows:

FIRST COST.

Boilers.....	\$ 3,500
Engines and generators.....	8,500
Heaters, pumps, piping, etc.....	4,000
Total.....	\$16,000

OPERATING COST.

Interest and depreciation, at 9 per cent..	\$1,440
Taxes, at 1.5 per cent.....	240
Repairs and supplies, ash removal, etc..	840
Additional labor.....	1,700
Water, at 5 cents per 1,000 gal.....	80
Total.....	\$4,300

Total kilowatt hours per year

$$\frac{(2,200 \times 6 \times 31) + (2,000 \times 6 \times 21)}{1.75} = 377,829$$

Operating cost per kilowatt hour, *not including coal*, $4,300 \div 377,829 = \$0.0114$, or a total, including coal, at \$3 per ton, of $0.114 + 0.0063 = \$0.0177$ per kilowatt hour.

392. *Effect of Compound Engines.*—it is quite possible in the above case that a saving could be made by using compound engines requiring 24 pounds of steam per I. H. P. per hour as against 30, for the simple engines, and costing, say \$5 per I. H. P. more. For simplicity we will assume that there will be sufficient exhaust for heating purposes throughout the year with either type of engine, which is practically the case, except for a very short time in the morning.

Total horse-power hours per year 661,200.

Saving in steam, $661,200 \times 6 = 3,967,200$ pounds.

Saving in cost of water

$$\frac{3,967,200}{8.3 \times 1,000} \times 0.05 = \$23.90.$$

Saving in cost of fuel, assuming an evaporation of 8.5 pounds of steam per pound of coal

$$\frac{3,967,200}{8.5} \times 0.0015 = \$700.09.$$

Difference in cost of engines, $341 \times 5 = \$1,705$. Additional for interest, depreciation and taxes, $1,705 \times 0.105 = \$179.03$.

The total saving is therefore $(23.90 + 700.09) - 179.03 = \$544.96$.

393. Effect of Condensing.—Whether or not there would be any saving in the present case by the use of a condenser for taking care of the surplus exhaust, especially in the summer time, is easily determined. Assuming that the condensation of each pound of steam requires 35 pounds of cooling water at summer temperatures, and that the water-rate of the engines will be reduced 15 per cent, the total weight per I. H. P. will be approximately $30 \times 0.85 \times 35 = 893$, or 900 pounds per hour in round numbers. The cost of this at 5 cents per 1,000 gallons will be

$$\frac{900}{8.3 \times 1,000} \times 0.05 = \$0.0054 \text{ per horse-power hour.}$$

The cost of fuel at \$3 per ton in case of the simple non-condensing engine is $\frac{30}{8.5} \times 0.0015 = \$0.0053$ per horse-power hour. If

15 per cent in cost of fuel is saved by running condensing, it will amount to only \$0.0008, which is about one-seventh the cost of condensing water. Hence, it is evident that it will be more economical to run non-condensing at all seasons of the year than to purchase the necessary quantity of water at city rates in this particular case. If cooling water could be obtained from a nearby lake or river without cost, except for pumping, the result might prove more favorable.

The above example has been worked out to show the methods of computation rather than actual quantities, although the assumptions have been reasonable and have been checked by figures reported in actual cases.

CHAPTER XI.

EQUIPMENT AND OPERATING COSTS.

In estimating the cost of all power and heating work quotations or bids for the actual equipment and construction should be obtained in each individual case when accurate results are desired. The data given in the following pages, while taken from reliable sources, is necessarily somewhat general in its nature and is to be used especially for filling in gaps when making estimates, and for preliminary approximations.

Cost of Power Equipment.

394. *Steam Power Plants.*—The cost of plant equipment varies considerably according to the size, type, and other local conditions. Table LXXXVI. has been made up from a series of curves prepared by Mr. W. C. Way and published in *Power*, March 17, 1908. These figures cover the steam equipment only. In case of an electric plant from \$14.00 to \$16.00 per horse power should be added for generators, foundations, and switchboard.

TABLE LXXXVI.

COST OF STEAM POWER PLANTS.

Size of plant, H.P.	Cost per brake or delivered horse power.		
	Steam plant equipment.	Building.	Total.
SIMPLE NON-CONDENSING ENGINES, RET. TUBULAR BOILERS.			
40	\$64	\$33	\$97
60	56	30	86
80	50	28	78
100	46	26	72
COMPOUND CONDENSING ENGINES, RET. TUBULAR BOILERS.			
200	\$50	\$22	\$72
300	46	18	64
400	43	16	59
500	42	14	56
COMPOUND CONDENSING ENGINES, WATER-TUBE BOILERS WITH RESERVE.			
300	\$56	\$18	\$74
500	50	14	64
700	47	13	60
1,000	45	12	57

Estimates for similar equipment given by H. Von Schon in the *Engineering Magazine*, May 1907, correspond very closely with the above.

Table LXXXVII., made up from data given by Mr. Reginald Trautschold in the *Practical Engineer*, January 1914, covers the entire equipment for steam power plants including *land* and building. It will be noted in comparison with Table LXXXVI., that the figures are considerably higher for the smaller plants, while for 1,000 horse-power capacity they are practically the same, although the cost of land is included.

Here, as before, the cost of generators and switchboard must be added in the case of electric plants.

TABLE LXXXVII.

COST OF STEAM POWER PLANTS.

Size of plant, H.P.	Cost per brake horse power, in- cluding land and building.
100	\$175
200	150
300	125
400	110
500	95
600	85
800	70
1,000	60
1,500	58

395. *Steam Electric Plants.*—Mr. F. W. Dean, in the *Practical Engineer*, October 1910, estimates the cost of a 1000-I. H. P. steam electric plant as \$69,750 for reciprocating engines and \$54,600 for a turbine plant of the same capacity. The buildings are estimated at \$20,000 in the first case and \$12,000 in the second.

396. *Isolated Plants.*—Table LXXXVIII. is taken from data presented by Mr. Percival R. Moses before the A. I. E. E., Jan. 12, 1912, and later published in the *National Engineer*.

The figures in this case are on a kilowatt basis and apply to the average isolated plant of medium size.

TABLE LXXXVIII.

COST OF MECHANICAL EQUIPMENT FOR ISOLATED PLANTS.

Kind of equipment.	Cost per kilowatt capacity.
<i>Boilers</i>	
Return tubular	\$14 - \$18
Water-tube	16 - 20
<i>Engines</i>	
High-speed simple non-cond. . .	20 - 25
Medium-speed comp. non-cond. .	28 - 35
Low-speed simple non-cond. . .	25 - 30
Low-speed comp. condensing. . .	20 - 25
<i>Generators</i>	
High-speed, direct-connected. . .	13 - 16
High-speed, belted	12 - 15
Low-speed, direct-connected. . .	16 - 20
<i>Switchboard</i>	5 - 10
<i>Foundations</i>	5 - 10
<i>Steamfitting</i> , including all auxiliary apparatus, piping, etc.	20 - 30

The figures in the above table are for all equipment erected and ready for use.

397. *Steam Equipment on Horse-power Basis.*—Table LXXXIX. is made up of average values taken from Table LXXXVIII., and gives the cost of steam equipment per horse power on a basis of 1.6 I. H. P. per kilowatt, which is approximately true for high and moderate speed engines of medium size. As the water-rate of the engines falls below 30 pounds per I. H. P. per hour, the cost of boilers, as given in the table, will evidently exceed the actual cost, but for the types of engines noted it will not be greatly in excess, and in any case is on the side of safety. For engines having water-rates below 25 pounds the cost of boilers should be corrected to correspond with the actual conditions when a close estimate is desired.

TABLE LXXXIX.

COST OF STEAM EQUIPMENT FOR ISOLATED PLANTS.

Kind of equipment.	Cost per indicated horse-power of engines.
<i>Boilers</i>	
Return-tubular	\$10
Water-tube	12
<i>Engines</i>	
High-speed simple non-cond. . .	14
Medium-speed comp. non-cond. .	20
Low-speed simple non-cond. . .	18
Low-speed comp. condensing. . .	14
<i>Foundations</i>	5
<i>Steamfitting</i> , including all auxiliary equipment, piping, etc.	16

Cost of Heating and Ventilating Equipment.

398. *Direct-Steam Heating.*—Here, as in power work, the cost will vary considerably in different cases, even in buildings of the same general type, but for purposes of approximation we may allow \$0.80 per square foot of direct radiation for dwelling houses, and \$1.00 per square foot for office buildings and similar work. In the case of return-line vacuum systems, allow \$1.15 per square foot. These figures are for the work erected and include all labor and material, except boilers, which are considered as part of the power equipment in combined power and heating plants.

399. *Forced Hot-Water Heating.*—The cost of installation in this case is slightly more than for vacuum heating and may be taken as \$1.25 per square foot of radiating surface which includes circulating pumps and heaters.

400. *Ventilation.*—The cost of ventilating equipment may be based on the volume of air moved per minute. This, however, will vary somewhat according to whether the fans are engine or motor driven and also upon the type of heating surface employed.

For average conditions, with motor-driven fans, the cost of equipment will run from \$75 to \$80 per 1,000 cubic feet of air per minute for a rise of 70 degrees in temperature. This includes fan, motor, heater, and air washer. If the washer is omitted, deduct \$25 from the above.

The cost of distributing ducts is difficult to estimate, especially in the case of office buildings, schoolhouses, etc., where conditions vary greatly. Galvanized iron work may be estimated on a basis of 14 to 16 cents per pound erected, allowing about 1 pound per 100 cubic feet of space for shops and factories with exposed overhead distributing pipes. In other buildings the weight of iron must be estimated from the plans.

Cost of Operation.

401. *Division of Costs.*—The cost of operation in case of a power and heating plant is divided into two parts, known as *fixed charges* and *operating costs*.

402. *Fixed Charges.*—These include investment costs, made up of interest, depreciation, rental, taxes, and insurance.

Administration costs, which occur in the case of central station work do not usually appear in connection with isolated plants.

403. *Operating Costs*.—These are made up of labor, fuel and water, oil, waste and supplies, repairs and maintenance, although in some cases the last item is included in the fixed charges.

404. *Load Factor*.—The relation between the capacity of a plant and the actual output is an important item in estimating the cost of power in any given case. This relation is commonly expressed in two ways, known as the *yearly load factor* and the *station load factor*.

$$\text{Yearly load factor} = \frac{A}{B \times 8,760}$$

in which

A = the yearly output, either in horse-power hours or kilowatt hours.

B = the rated capacity, either in horse power or kilowatts.

8,760 = the number of hours in a year, (365×24).

$$\text{Station load factor} = \frac{A}{B \times C}$$

in which

A and B are the same as above.

C = the hours per year the plant is in actual operation.

An inspection of the items comprising the fixed charges makes it evident that this part of the operating cost is not affected by the load factor, as these items, with the possible exception of depreciation, remain constant regardless of whether the plant is running or not.

The cost per unit of power, however, varies inversely as the load factor, first because all parts of the equipment are more efficient when operated at or near their normal rating, and second because the amount of idle capital invested decreases as the load factor increases.

The total operating cost increases with the load factor, due to the increase in fuel and other supplies, but not in direct proportion.

405. *Interest*.—The rate of interest varies in different localities and at different times, but where no specific information is at hand it is customary to assume 5 per cent on mortgages secured by this class of property.

406. *Depreciation*.—This represents an amount which must be laid aside yearly to replace the equipment as it wears out or becomes obsolete. It is often a difficult matter to estimate this with any degree of accuracy as so much depends upon the design of the plant, the quality of the equipment and the care which it receives. In making approximations it is usually safe to allow 5 per cent on the first cost of the plant for this item.

407. *Taxes*.—This is a variable factor, usually running from 1 to 2 per cent of the first cost of the equipment. A common method, when exact data is lacking, is to allow a charge of 2 per cent on one-half to three-fourths the first cost for new plants and those which have been kept in good repair, which is evidently the same as 1 to 1.5 per cent on the full cost.

408. *Insurance*.—Buildings and machinery are commonly insured against fire loss, and boilers against explosion, at an annual charge running from one-half to 1 per cent of the value of the property.

409. *Labor*.—The item of labor in the case of an isolated plant is usually a fixed charge so far as actual results are concerned, although included in the operating costs.

The cost of labor depends largely upon the locality and the conditions under which the plant is operated.

When no definite information is at hand the wages of a day engineer may be estimated at \$4 per day, night engineer \$3, firemen and helpers \$2 per day.

410. *Fuel and Water*.—The cost of fuel and water must be estimated in each particular case from local prices, and will of course vary with the size and efficiency of the plant and the conditions under which it is operated. Data for determining the quantity of fuel for power purposes has already been given in Paragraphs 205, 226 and 227.

411. *Oil, Waste and Supplies*.—Estimates for these items, given by Mr. Wm. O. Webber in the Engineering Magazine, July 1908, based on compound condensing engines are as follows:

TABLE XC.

COST OF OIL, WASTE AND SUPPLIES.

Size of plant, H.P.	Cost of oil, waste and supplies per horse power per year, based on a 10-hour day.
100	\$2.40
200	2.00
400	1.45
600	1.08
800	0.83
1,000	0.70
1,500	0.65

412. *Repairs and Maintenance.*—This covers the cost of keeping the plant in running order and depends to a large extent upon the age of the equipment, the care which it receives and the service required.

Under ordinary conditions a charge of 2 to 3 per cent of the first cost per year will cover this item.

413. *Cost of Heating.*—The cost of heating depends upon the amount of radiating surface in the building, the care used in regulation, the length of the heating season and whether or not heat is shut off at night, and the cost of fuel.

Taking the heating season in New England and the Northern States as seven months, it will amount to approximately 5,000 hours of continuous heating or 3,300 hours if the radiation is shut off for 8 hours during the night.

As a comparatively small amount of fuel is saved by allowing the building to cool off at night, when "warming up" in the morning is taken into consideration, it is usually safer to count upon continuous warming when estimating the cost of heating. If the radiation is shut off at night the hours of heating should not, in general, be taken less than 4,000 per year. Each square foot of direct radiation gives off approximately 250 B. T. U. per hour for steam, and 210 B. T. U. for hot water under forced circulation. Therefore, the total B. T. U. required per year with the radiators working at their full capacity is given by the expression, $H \times R \times E$, in which,

H = hours the radiation is in use.

= 5,000 for continuous warming.

= 4,000 when radiation is shut off for seven or eight hours at night.

R = total radiating surface.

= square feet of surface in radiators $\times 1.2$.

E = efficiency of radiating surface.

= 250 for direct steam.

= 210 for direct hot water under forced circulation.

This, as above stated, is for the full capacity of the radiation.

For different methods of regulation multiply the results given by the above by the following factors, taken from Paragraph 208.

Method of regulation.	Factor.
Low-pressure steam; hand regulation.....	0.80
Vacuum system; graduated valves.....	0.70
Forced hot water; hand regulation.....	0.60
Steam or hot water; automatic control.....	0.55

Having determined the heat units to be supplied, the approximate cost may be found by the formula:

$$C = \frac{U \times T}{16,000,000}$$

in which

C = total cost of fuel.

U = total heat units required.

T = price of fuel per ton.

414. *Cost of Ventilation*.—The B. T. U. required for ventilation may be found by the expression:

$$H \times V \times 0.64$$

in which

H = total number of hours ventilation is provided during the heating season.

V = cubic feet of air supplied per hour.

In the above, an average outside temperature of $+35$ degrees has been assumed, which corresponds well with New England conditions.

As the temperature of the entering air may be closely regulated by the engineer, no correction to the above is necessary.

Having determined the total heat units to be supplied the fuel cost is computed the same as for heating.

The item of power for driving the fans, being small in comparison with the fuel cost, has been omitted. If, however, mechanical ventilation is provided during the non-heating season also, it should be included.

415. *Total Cost.*—In computing the total cost of operation the fixed charges must first be estimated for the entire plant, including power, heating, and ventilating equipment.

Next the various items making up the operating cost must be considered, charging each to its respective section of the plant, taking into account the utilization of exhaust steam as already described.

While general data relating to the cost of power may be of value in the case of central stations and in plants where the exhaust steam is thrown outboard or condensed, it is of little use in connection with isolated plants.

Here the most important factor is usually the utilization of exhaust steam which varies widely in different cases according to local conditions.

Tables XCI. and XCII., however, from data given by Mr. Reginald Trautschold in *Practical Engineer*, Jan. 1, 1914, cover items which may often be used to advantage in making approximations of the cost of power, even in problems involving the use of exhaust steam for heating. The figures given in Table XCI. include the items of attendance, oil, waste and supplies, and are based upon average prices.

TABLE XCI.

COST OF ATTENDANCE, OIL, WASTE AND SUPPLIES.

Size of plant, brake H.P.	Cost of attendance, oil, waste and supplies per 10-hour day, based on 300 working days, per year.
100	\$ 4.85
200	8.00
400	11.60
600	13.00
800	13.50
1,000	14.20
1,500	19.50

Table XCII. gives the tons of coal burned per day in plants of different size, from which the fuel cost may be computed by using current prices. It will be noticed from an inspection of the table that the amount of coal per unit of power delivered varies inversely as the size of the plant. This is due not only to the higher efficiency of the boilers but also to the lower water-rates

of engines and turbines and the higher mechanical efficiency obtained in plants of larger size.

TABLE XCII.

AMOUNT OF COAL REQUIRED PER YEAR.

Size of plant, brake H.P.	Tons of coal required per 10-hour day, based on 300 working days per year.
100	3.3
200	6.0
400	10.7
600	12.4
800	12.8
1,000	13.1
1,500	14.0

CHAPTER XII.

PRACTICAL EXAMPLES IN DESIGN AND COST OF OPERATION.

In the previous chapters special attention has been given to the determination of heat and power requirements, available exhaust, etc., while the matter of cost, as regards plant equipment, and operation, has been approximated rather than accurately determined. In the present chapter this is reversed, steam quantities and general conditions being approximated to some extent, while changes in plant design are taken up in some detail and costs more carefully estimated. In this way each feature of the problem is considered, both from an accurate and an approximate standpoint, which is of considerable value in this class of work. Often all that is required at first is a preliminary examination and report to show in a general way what may be expected from different arrangements and methods of operation, either as to steam supply and demand, or plant costs. This, in many cases, will answer as well as more elaborate computations and will prove less expensive for the client. On the other hand, conditions may be such as to require more accurate results, either for comparison with central station rates or for use in plant design.

The following examples are worked out along the lines above noted, taking up different types of buildings to be constructed, or already in use, showing changes in cost of construction and operation when generating their own power under varying conditions.

416. Purpose of the Problems.—As previously stated, these problems are for purposes of illustration rather than for giving accurate figures as to the cost of plant equipment and the production of power. At the same time they will closely approximate the results which may be expected under the particular conditions assumed and will be found useful in practical work by substituting actual values and by extending their scope to include other factors not mentioned.

Problem (1)—Technical High School.

417. *Proposition to be Considered.*—First, take the case of a technical high school, to be built in the residential part of a city where the ordinance requires that only anthracite coal or coke shall be burned. Power is required for driving the ventilating fans, for lighting, and for operating the machinery in the manual training department. The question is, shall a low-pressure steam plant be installed for heating and ventilating the building, and power be purchased from a local company at 4 cents per kilowatt hour, or shall the necessary current be generated upon the premises?

418. *General Requirements.*—Let the general requirements be as follows: The space to be heated and ventilated is equivalent to 40 class rooms of standard size. The entire building is to be in use for class-room work 6 hours in the daytime, for 5 days in the week, for a school year of 36 weeks. Evening classes are to be held in 20 class rooms for 3 hours each day during the same period. Ventilating fans are to be started $\frac{1}{2}$ hour before classes assemble, to flush out the building, and are to be run continuously throughout the sessions. This makes $6\frac{1}{2}$ hours in the day time and $3\frac{1}{2}$ hours in the evening. The necessary lights are to be burned 7 hours for the day session and 4 hours for the evening session. Day lighting is comparatively small in amount and relates to certain corridors, toilets, basement rooms, etc. Power for machine work is required 4 hours during the day session and 3 hours in the evening.

419. *Heat Requirements.*—A standard class room requires approximately 40,000 B. T. U. per hour for heating purposes in zero weather, when the building is of ordinary construction and the proportion between wall and window surface about the average.

Assuming 50 pupils per class room, an air supply of 40 cubic feet per pupil per minute, will call for $50 \times 40 = 2,000$ cubic feet of air per minute, or $2,000 \times 60 = 120,000$ per hour. To raise this from 0 to 70 degrees temperature will require $120,000 \times 1.3 = 160,000$ B. T. U. in round numbers, making a total of $160,000 + 40,000 = 200,000$ B. T. U. per hour per room.

This calls for a maximum of $200,000 \times 40 = 8,000,000$ B. T. U. per hour in the day time and $200,000 \times 20 = 4,000,000$ B. T. U. in

the evening. The entire system is to be equipped with automatic temperature control so the actual heat required will be nearly proportional to the difference between the inside and outside temperatures, as already described.

420. Boiler Power for Heating.—One boiler horse power represents a capacity of approximately 30,000 B. T. U. per hour; hence, the maximum to be provided for heating and ventilating purposes is $8,000,000 \div 30,000 = 267$ horse power.

In the evening the maximum power required will be $4,000,000 \div 30,000 = 134$ horse-power.

Horizontal tubular boilers will be the best type for a low-pressure heating plant of this size, using 4 units of 67 horse power each, set in one battery. The full number will then be required in the daytime, while 2 will be sufficient for evening use. In the mild weather of spring and fall, one or more can be used as found necessary. Suppose the weather records in this locality show the average day temperature for the heating season to be $+40$ degrees, and the evening temperature $+30$ degrees. The factors for these temperatures as found by curve No. 2, Fig. 211, are 0.48 and 0.61, respectively, making the average boiler power for day service $267 \times 0.48 = 128$ horse power and for evening $133 \times 0.61 = 81$ horse power.

421. Ventilating Fans and Motors.—The total air volume to be handled during the day session is $2,000 \times 40 = 80,000$ cubic feet per minute. As only half this quantity will be required in the evening, it will be best to use two fans and two motors, each unit having a capacity of 40,000 cubic feet per minute. We find from Paragraphs 131 and 132, that a 9-foot fan running at about 120 r. p. m. will furnish this amount of air and call for 15 D. H. P. This will require 30 D. H. P. for the day session and 15 D. H. P. for the evening.

422. Lighting.—Knowing the character and use of the different rooms, it is estimated that 6 kilowatts will be required for lighting during the day session and 30 kilowatts during the evening.

423. Power for Machine Driving.—This for the number and character of the machines to be driven is estimated at 40 D. H. P. As the machines are motor-driven, corrections must be made for the loss in efficiency on this account.

424. *Cost of Current.*—Before estimating the cost of current to be purchased, the kilowatt hours must be determined. Assuming average motor efficiencies of 75 per cent, they being small in most cases, and taking the ratio of horse power to kilowatt as 0.746 we have the following:

425. *Day Service.*

Fan motors—

D. H. P. = 30.

Hours in operation = $6\frac{1}{2}$.

$$30 \times 0.746 \times 6.5$$

$$\hline = 194 \text{ kilowatt hours.}$$

0.75

Lighting—

Kilowatts = 6.

Hours of service = 7.

$$6 \times 7 = 42 \text{ kilowatt hours.}$$

Machine shop—

D. H. P. = 40.

Hours in operation = 4.

$$40 \times 0.746 \times 4$$

$$\hline = 159 \text{ kilowatt hours.}$$

0.75

Total for day session $194 + 42 + 159 = 395$ kilowatt hours.

426. *Evening Session.*

Fan motors—

D. H. P. = 15.

Hours in operation = $3\frac{1}{2}$.

$$15 \times 0.746 \times 3.5$$

$$\hline = 52 \text{ kilowatt hours.}$$

0.75

Lighting—

Kilowatts = 30.

Hours of service = 4.

$$30 \times 4 = 120 \text{ kilowatt hours.}$$

Machine shop—

D. H. P. = 40.

Hours in operation = 3.

$$40 \times 0.746 \times 3$$

$$\hline = 120 \text{ kilowatt hours.}$$

0.75

Total for evening session $52 + 120 + 120 = 292$ kilowatt hours.

Total for day $395 + 292 = 687$ kilowatt hours.

427. *Cost of Current.*—This at 4 cents per kilowatt hour will amount to: $687 \times 0.04 = \$27.48$ per day, or $27.48 \times 5 \times 36 = \$4,946.40$ for the school year.

428. *Private Power Plant.*—Let us now see what additional equipment is necessary for generating the current on the premises. We will assume that water-tube boilers will be substituted for fire-tube on the score of safety, although the latter are used

for power purposes in many public buildings. It will also be necessary to add something to the boiler power to make up for the steam lost in passing through the engine, even though all of the exhaust is utilized for heating purposes. In previous computations this has been taken as 15 per cent, and the same will be assumed here.

There will be no other change in the heating system except the introduction of a pressure reducing valve, a back pressure valve, and an automatic pump and receiver for returning the condensation to the boilers against a higher pressure. This arrangement will also call for separate traps on the returns from the direct radiation and the main heaters.

No change is required in the motors or in the wiring, except connections between the generator and the switchboard.

429. *Additional Equipment.*—The principal items to be included in new or additional equipment are as follows:

Boiler.	Separator.
Engine.	Valves and Exhaust Head.
Generator.	Traps.
Pump and Receiver.	Piping.
Feed-water Heater.	

430. *Engine and Generator.*—The maximum power requirements at the generator during the day are as follows:

Fan Motors.....	30×0.746	
		= 30 kilowatts.
	0.75	
Machine Motors.....	40×0.746	
		= 40 kilowatts.
	0.75	
Lighting.....		.6 kilowatts.
Total.....		.76 kilowatts.

Maximum requirements during the evening:

Fan Motors.....	15×0.746	
		= 15 kilowatts.
	0.75	
Machine Motors.....	40×0.746	
		= 40 kilowatts.
	0.75	
Lighting.....		.30 kilowatts.
Total.....		.85 kilowatts.

Assuming an engine capacity of 1.75 I. H. P. per kilowatt, it calls for $76 \times 1.75 = 133$ I. H. P. at the generator engine for the day load, and $85 \times 1.75 = 149$ I. H. P. for the evening load, which may be taken as 150 in round numbers.

In order to allow for a slight increase in power we will base our estimate on a 90-kilowatt generator and a 160-horse power simple high-speed engine having a water-rate of 30 pounds of steam per I. H. P. per hour at full load.

431. Boiler Capacity for Power and Heating.—The maximum boiler power will be required during the daytime when the entire building is in use. Assuming 15 per cent loss in the steam used by the engine, and a water-rate of 32 pounds per I. H. P. per hour at the load carried (about 84 per cent of full load), the "make-up" water will amount to $133 \times 32 \times 0.15 = 638$ pounds, calling for $638 \div 30 = 21$ boiler H. P., thus making the total maximum boiler capacity required $267 + 21 = 288$ H. P.

432. Size of Pumps, Heater, Etc.—The receiving tank for a plant of this size should be about 36 inches in diameter by 48 inches in length. It should be connected with two 6-inch $\times$ 4-inch $\times$ 6-inch boiler feed pumps so arranged that they may be used either separately or together, each being of sufficient size to do the entire work when running at moderate speed.

The extra pump is to serve as a relay in case of repairs.

A feed-water heater should be provided for use when all, or a portion, of the exhaust is thrown outboard in mild weather. and should have a capacity of $\frac{160 \times 30}{30} = 160$ H. P. for maximum conditions.

433. General Arrangement of Power Apparatus and Piping.—In Fig. 213 the piping arrangement is shown for a simple low-pressure heating system, with the condensation returning to the boilers by gravity. Fig. 214 illustrates the piping required by the combined power and heating plant. Live steam enters the heating main through a pressure reducing valve provided with a by-pass and cut-out valves. The high-pressure supply to the engine is taken from the main drum through an independent connection. The exhaust pipe connects with the heating main through an oil separator, and also has a branch leading outboard through a back

pressure valve. The feed-water heater is on a by-pass connected with the main exhaust pipe, and so valved that it may be used or not as desired. The condensation from both the direct and indirect systems of heating is trapped to the receiving tank as shown, and returned to the boilers through the feed pipe. Fresh feed water from the heater is drawn into the tank as needed and pumped into the boilers with the condensation.

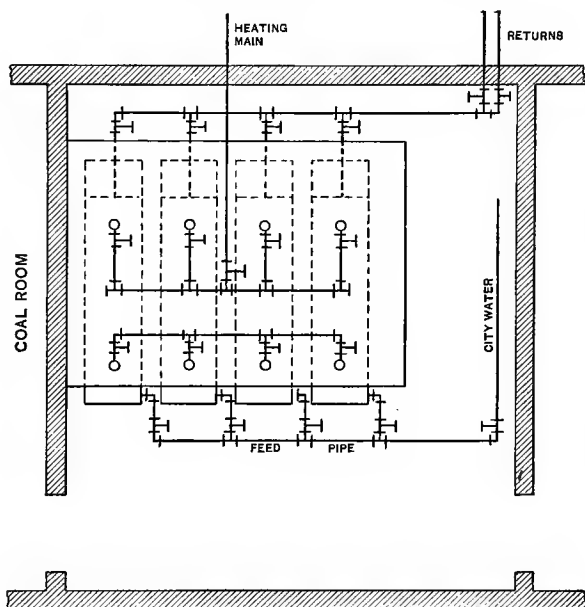


Fig. 213. Low-Pressure Heating Plant for Technical High School.

434. *Cost of Power Equipment.*—This will vary somewhat in different localities, but the following may be taken as a fair average for the apparatus installed ready for use:

Engine and dynamo, including foundation.	\$4,000
Difference in cost of boilers.....	1,000
Piping, tanks, pump, feed-water heater, etc.	2,000
Total.....	\$7,000

435. *Cost of Operation.*—The cost of operation will depend upon the proportion of exhaust steam which can be utilized in

the heating system, and will, of course, vary with the outside temperature.

Steam required by the engine at maximum *day* load is $133 \times 32 = 4,256$ pounds per hour. Lost in work and cylinder wastes, $4,256 \times 0.15 = 638$ pounds per hour.

Available exhaust for heating purposes, $4,256 - 638 = 3,618$ pounds per hour. Steam required per hour for heating and ventilation with an outside temperature of zero,

$$\frac{200,000 \times 40}{960} = 8,333 \text{ pounds.}$$

When the outside temperature reaches $+44$ degrees, the steam requirements for heating and ventilation become $8,333 \times 0.43 =$

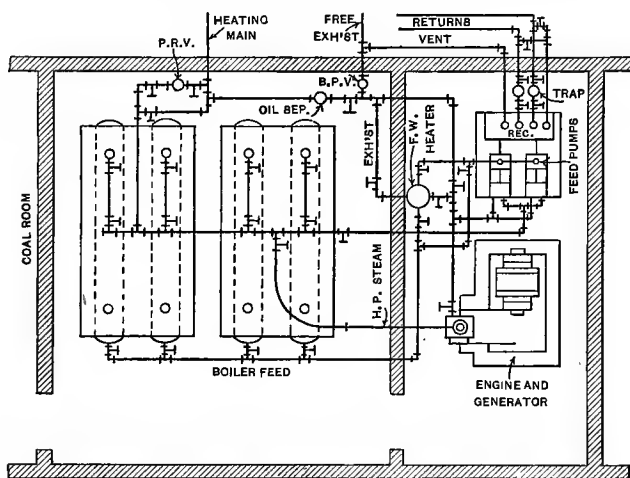


Fig. 214. Combined Power and Heating Plant for Technical High School.

3,583 pounds per hour, which is practically the same as the available exhaust.

(The factor 0.43 is from Fig. 211, Curve 2, for a temperature of 44 degrees.)

Hence, for outside temperatures below $+44$ degrees the total fuel cost per hour for power purposes during the day will be simply that necessary to replace the 638 pounds of steam lost in work and cylinder wastes, and which, with an evaporation of

8 pounds of steam per pound of coal, and coal at \$6 per ton, will amount to

$$\frac{638 \times 0.003}{8} = \$0.239.$$

At +50 degrees the steam requirements for heating and ventilation amount to $8,333 \times 0.36 = 3,000$ pounds per hour, and the waste will be $(3,618 - 3,000) + 638 = 1,256$ pounds, at a cost of

$$\frac{1,256 \times 0.003}{8} = \$0.471.$$

In a similar manner, the fuel cost per hour for different outside temperatures is found to be as follows:

TABLE XCIII.

FUEL COST FOR POWER AT MAXIMUM DAY LOAD.

Outside temperature, degrees.	Fuel.	
	Cost per hour (coal at \$6 per ton).	Cost, per kw.-hour.
+44 (and below).....	\$0.239	\$0.00315
+50.....	0.471	0.00620
+55.....	0.689	0.00906
+60.....	0.877	0.01154
+65.....	1.096	0.01442
+70.....	1.284	0.01689

During the *evening* session, 150 I. H. P. are required, and, being practically full load, we will assume a water rate of 30 pounds per I. H. P., making a total of $150 \times 30 = 4,500$ pounds per hour. Of this the loss in work and cylinder waste amounts to $4,500 \times 0.15 = 675$ pounds, leaving $4,500 - 675 = 3,825$ pounds of exhaust steam per hour available for heating purposes.

The steam required per hour for heating and ventilation with an outside temperature of zero is

$$\frac{200,000 \times 20}{960} = 4,167 \text{ pounds.}$$

At +6 degrees the steam requirements for heating and ventilation become $4,167 \times 0.92 = 3,834$ pounds per hour which is practically equal to the available exhaust. From the above conditions Table XCIV. has been prepared in a similar manner to Table XCIII.

TABLE XCIV.

FUEL COST FOR POWER AT MAXIMUM EVENING LOAD.

Outside temperature, degrees.	Fuel.	
	Cost per hour (coal at \$6 per ton).	Cost per kw.-hours.
+ 6 (and below).....	\$0.253	\$0.00298
+10.....	0.328	0.00386
+20.....	0.531	0.00625
+30.....	0.734	0.00864
+40.....	0.938	0.01104
+50.....	1.125	0.01324
+60.....	1.328	0.01562
+70.....	1.531	0.01800

In making an estimate of yearly costs for comparison, it is necessary to divide the school season into two parts; one representing the time when all of the available exhaust may be used in the heating system and the only fuel cost for power is represented by the steam lost in work and cylinder wastes, and the other, when the outside temperature is above +44 degrees in the daytime and above +6 degrees in the evening.

Suppose the weather records for previous years show that during the school year there have been 18 weeks when it has been below +44 degrees during the day sessions, and that the average for the remaining 18 weeks has been +60 degrees. Also that during the evenings there have been 4 weeks when the temperature has been below +6 degrees, and that for the remaining 32 weeks it has averaged +50 degrees during that time.

Taking the *day* sessions we have $18 \times 5 = 90$ days, when the fuel cost is \$0.00315 per kilowatt hour (from Table XCIII.). The total kilowatt hours for a day session is 395, hence, the total fuel cost for 90 days is $90 \times 395 \times 0.00315 = \$111.98$. For the remaining weeks, with an average outside temperature of +60 degrees, the cost per kilowatt hour (Table XCIII.), is \$0.01154, giving a total of $18 \times 5 \times 395 \times 0.01154 = \$410.25$.

For the *evening* sessions we have for 4 weeks a fuel cost of \$0.00298 per kilowatt hour (Table XCIV.), amounting to $4 \times 5 \times 292 \times 0.00298 = \$17.40$, and for the remaining 32 weeks, $32 \times 5 \times 292 \times 0.01324 = \$618.57$. Therefore, the total fuel cost, for power alone, for the school year will be $111.98 + 410.25 + 17.40 + 618.57 = \$1,158.20$.

436. Comparison of Costs.—In making a comparison with the cost when current is purchased from outside, various items must be taken into account, as listed below:

Interest and depreciation on investment at 10 per cent.	\$ 700.00
Taxes, 2 per cent.	140.00
Repairs and supplies.	250.00
Additional labor.	500.00
Additional water.	60.00
Cost of coal.	\$1,650.00
Total.	\$2,808.20
Cost of current when purchased.	\$4,946.40
Cost of current when generated.	
	2,808.20
Saving for the school year.	
	\$2,138.20

In estimating the labor, as above, it has been assumed that no additional men will be required, but that a higher priced man will be necessary for chief engineer.

Problem (2)—Hotel.

437. General Proposition.—The present example takes up the case of a large hotel requiring steam for heating, ventilating, hot-water service, and laundry work; also power for lighting, refrigeration, the driving of ventilating fans and laundry machinery, and elevator service. At present the plant consists of a battery of low-pressure boilers for heating on a vacuum system, a high-pressure boiler for operating the refrigerating machine, the laundry engine, and also for furnishing high-pressure steam for the laundry mangles. Electric current for operating the elevators, and fan motors, and for lighting the building, is purchased from a central plant at the rate of 2 cents per kilowatt hour.

It is proposed to remove part of the old boilers and install new high-pressure boilers of the water-tube type, and equip the entire plant with mechanical stokers for burning a grade of coal which may be purchased for \$3 per ton. It is also proposed to install engines and dynamos and generate the current for operating the various motors and for lighting the building. Required to determine what changes in plant equipment are necessary and compute the cost of same, also determine so far as pos-

sible, the saving in operating expenses, assuming the various requirements for steam and power to be as follows:

438. Heating and Ventilation.—The building contains 26,700 square feet of direct steam radiation requiring $26,700 \times 250 \times 1.2 = 8,000,000$ B. T. U. per hour in round numbers, in zero weather. The ventilation of the kitchen, dining-rooms, office, and other public rooms calls for an air supply of 8,000 cubic feet per minute, requiring $8,000 \times 60 \times 1.3 = 624,000$ B. T. U. per hour.

439. Hot-Water Service.—The total hot water to be furnished, including that for laundry and baths, amounts to 18,000 gallons in 24 hours. A large storage tank is provided so that boiler capacity for heating the entire supply in 10 hours is all that is necessary. It requires about 1,100 B. T. U. to raise the temperature of 1 gallon of water from 50 to 180 degrees, hence $1,800 \times 1,100 = 1,980,000$ B. T. U. per hour are required for hot-water service.

440. Refrigeration.—The refrigerating plant is equipped with a 15-ton machine driven by a steam engine using 34 pounds of steam per I. H. P. This calls for $15 \times 1.4 \times 34 = 714$ pounds of steam per hour. (See Paragraph 195.)

441. Laundry Machines.—The laundry engine is rated at 20 I. H. P. and requires 40 pounds of steam per I. H. P. per hour, or a total of $20 \times 40 = 800$ pounds.

The mangles require approximately 400 pounds of high-pressure steam per hour.

442. Electric Plant.—The various fan motors and those for the kitchen machinery are to supply a total of 14 D. H. P., requiring $14 \div 0.7 = 20$ I. H. P. at the generator engine.

The lighting plant requires a maximum of 80 kilowatts, or an I. H. P. of $80 \times 1.75 = 140$ at the generator engine.

Four electric elevators, each having a floor space of 64 square feet, are in practically constant use at about 70 per cent of their rated capacity throughout the day, calling for

$$\frac{64 \times 4 \times 0.35 \times 0.7}{0.7} = 90 \text{ I. H. P. at the engine.}$$

(See Paragraph 194.)

The total I. H. P. required of the engine driving the generator is therefore $20 + 140 + 90 = 250$. For a simple high-speed engine this will require $250 \times 30 = 7,500$ pounds of steam per hour.

443. *Boiler Power.*—The total heat required per hour under maximum conditions for warming the building, ventilation and hot-water service, is $8,000,000 + 624,000 + 1,980,000 = 10,604,000$ B. T. U. This calls for $10,604,000 \div 960 = 11,042$ pounds of steam, and requires $11,042 \div 30 = 368$ boiler H. P.

For power work, the refrigerating machine calls for 714 pounds of steam per hour, the laundry engine 800 and the generator engine 7,500, making a total of $714 + 800 + 7,500 = 9,014$ pounds. To this must be added 400 pounds for the laundry mangles, which calls for a total of

$$\frac{8,014 + 400}{30} = 314 \text{ H. P.}$$

of high-pressure boilers. The exhaust steam available for heating purposes will be approximately $9,000 \times 0.85 = 7,650$ pounds per hour, or the equivalent of

$$\frac{7,650}{30} = 255 \text{ boiler H. P.}$$

This makes it necessary to retain $368 - 255 = 113$ H. P. of the present heating boilers.

The old plant consists of one 75-H. P. high-pressure boiler and five low-pressure boilers of the same capacity. It will be best, probably, to remove the high-pressure boiler and three of the heating boilers, replacing them with 314 H. P. of water-tube boilers designed for high-pressure service. By this arrangement the entire power and heating work may be done for a considerable part of the time by the new boilers, the two older ones being used only as needed for heating in the coldest weather.

444. *Engines and Generators.*—The total I. H. P. of the main engine was found to be 250, which calls for a generator of $250 \div 1.75 = 143$ kilowatt capacity. A more direct method of determining the size of generator is as follows: Assume efficiencies of 85 and 90 per cent for the fan and elevator motors respectively. Then we have required

$$\frac{14 \times 0.746}{0.85} = 12 \text{ kilowatts for fan motors, and}$$

$$\frac{64 \times 4 \times 0.35 \times 0.746 \times 0.7}{0.9} = 52 \text{ kilowatts for elevators}$$

making a total of $12+52+80 = 144$ kilowatts, which is practically the same as before. It will be best to install one 100-kilowatt set and two 50-kilowatt sets; the larger having a 175-H. P. engine and each of the smaller ones a 90-H. P. engine. This will give a spare machine, which is a matter of much importance in a plant of this size.

The larger unit, or both of the smaller ones, will give the required power during the daytime, while the larger one and one of the smaller ones will be necessary in the evening. During the night one of the smaller units will be sufficient for all purposes.

445. Cost of Changing Plant.—In estimating the cost of changing over the plant, we will assume that the old boilers are in good condition and that \$5 per horse power may be realized upon them after paying for the cost of removal. There are four 75-H. P. boilers to be removed (3 low-pressure and 1 high-pressure), which will amount to $4 \times 75 \times 5 = \$1,500$.

The new equipment installed will cost approximately as follows:

Boilers and stokers.....	\$5,300
Engines.....	4,500
Generators.....	3,000
Foundations.....	1,500
Piping, heater, receiver and pumps.....	4,700
Total.....	\$19,000
Less price of old boilers.....	1,500
Net cost.....	\$17,500

446. Lay-Out of Plant.—The general arrangement of the original plant is shown in Fig. 215 and the remodeled plant in Fig. 216. In the latter case the two low-pressure boilers connect directly with the heating main, while live steam from the high-pressure boilers is fed into the heating system through a reducing valve as needed. High-pressure steam is supplied to the various engines from a main header as indicated. The exhaust from the laundry engine is practically all utilized, partly in the dryer and partly for heating water.

The refrigerating machine runs continuously day and night and the exhaust is utilized for water heating, any surplus going into the general heating system of the building which is

operated on the vacuum plan. The exhaust from the generator engines, after being purified of oil, is turned into the heating main, there being an outboard branch provided with a back-pressure valve which opens automatically when the exhaust exceeds that required by the heating system. The feed-water heater is located, as shown, and is provided with a by-pass for use when the heater is not in operation. The condensation from the heating system, hot-water tank, etc., is either pumped or trapped to a vented receiver and pumped back to the boilers.

447. *Additional Fuel for Power.*—In estimating the additional fuel required for power purposes a different method has been

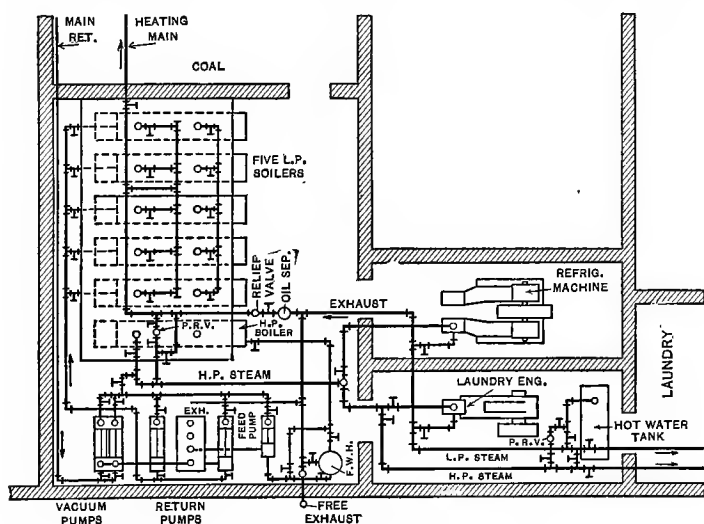


Fig. 215. Hotel—Original Plant.

employed from any yet given. First, the steam required for hot-water heating drops out of the problem, because this work is practically the same as in the old system. That is, all water heating is done by the exhaust from the laundry and refrigerating engines. The latter runs all night, and the exhaust is utilized for heating a large storage tank of water for use during the next morning. During the day the laundry engine is in use, and its exhaust makes up the additional amount needed. There is a live

steam connection with the high-pressure boilers through a reducing valve, and an exhaust connection to the heating system through a relief valve, for keeping the proper pressure on the heating coil in the storage tank. Assume for this particular locality a heating season of 210 days, and divide it into 7 periods of 30 days each. Separate each day into 3 periods of 8 hours each, which will represent approximately *day*, *evening* and *night*. This arrangement would not, of course, apply to all kinds of buildings, but for a large city hotel is fairly accurate. Obtain from the weather records of preceding years the average day,

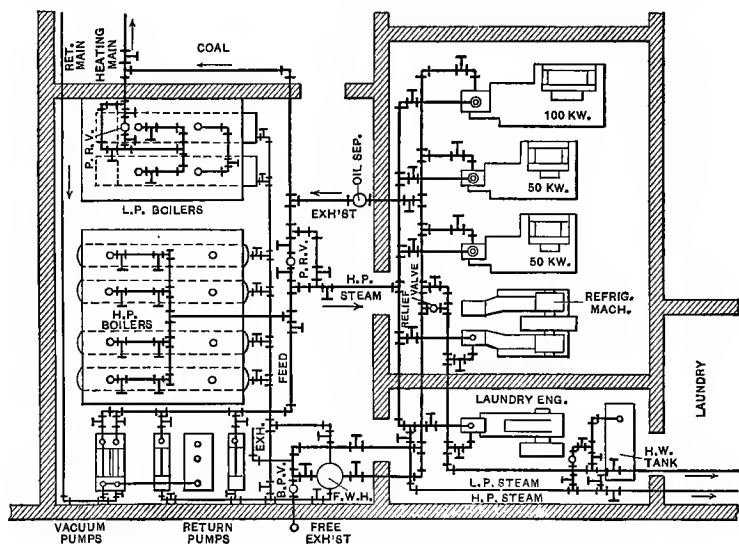


Fig. 216. Hotel—Remodeled Plant.

evening and night temperatures of these 7 periods, which extend from about the middle of October to the middle of May.

Let these be as shown in column 3, Table XCV.; the upper number in each group being the average day temperature, the second the evening temperature and the third the night temperature. Column 4 gives the factor for the given temperature as taken from curve No. 4, Fig. 211, which is for vacuum heating. Column 5 gives the pounds of steam required for heating and ventilating purposes for one hour in each 8-hour period, it being

TABLE XCV.

1 Time of year.	2 Period of day.	3 Average temperature, degrees.	4 Factor for the given temperature.	5 Pounds of steam per hour for heating.	Exhaust wasted.		
					6 Per hour, pounds.	7 In 1 hour of each period (3 hours).	8 In 30-day period, pounds.
October and November.....	Day.....	50	0.57	5,130	800		
	Evening.....	47	0.60	5,400	1,100		
	Night.....	40	0.66	5,511	225	2,125	510,000
November and December.....	Day.....	40	0.66	5,940	800		
	Evening.....	30	0.74	6,660	1,000	2,025	486,000
	Night.....	25	0.79	6,596	225		
December and January.....	Day.....	30	0.74	6,660	800		
	Evening.....	20	0.83	7,470	1,000	2,025	486,000
	Night.....	15	0.87	7,264	225		
January and February.....	Day.....	20	0.83	7,470	800		
	Evening.....	15	0.87	7,830	1,000	2,025	486,000
	Night.....	10	0.91	7,598	225		
February and March.....	Day.....	40	0.66	5,940	800		
	Evening.....	35	0.70	6,300	1,000	2,025	486,000
	Night.....	25	0.79	6,596	225		
March and April.....	Day.....	55	0.53	4,770	800		
	Evening.....	50	0.57	5,130	1,370	2,395	574,800
	Night.....	40	0.66	5,511	225		
April and May.....	Day.....	65	0.44	3,960	1,340		
	Evening.....	60	0.49	4,410	2,090	3,655	877,200
	Night.....	55	0.53	4,425	225		
Total.....							3,906,000

assumed that the building is warmed for the entire 24 hours and ventilated during the day and evening (16 hours). These quantities are based on a steam consumption of

$$\frac{8,000,000 + 624,000}{960} = 9,000 \text{ pounds per hour,}$$

in round numbers, for both heating and ventilation in zero weather, and $9,000 - 650 = 8,350$ pounds for heating alone.

Column 6 gives the pounds of steam wasted by the generator engines for one hour in each period.

In computing the exhaust steam the following power requirements are assumed for each 8-hour period:

448. *Day Load.*—

	I.H.P.
Lighting.....	65
Elevators.....	90
Fan motors, etc.....	20
	175

Available exhaust $175 \times 30 \times 0.85 = 4,500$ pounds (approx.).

Lost in work and cylinder wastes, 800 pounds (approx.).

449. *Evening Load.*—

	I.H.P.
Lighting.....	140
Elevators.....	55
Fan motors, etc.....	20
	215

Available exhaust $215 \times 30 \times 0.85 = 5,500$ pounds (approx.).

Lost in work and cylinder waste, 1,000 pounds (approx.).

450. *Night Load.*—

	I.H.P.
Lighting.....	40
Elevators.....	10
	50

Available exhaust $50 \times 30 \times 0.85 = 1,275$ pounds.

Lost in work and cylinder waste, 225 pounds.

Whenever the available exhaust is less than the heating requirements for the corresponding period, the exhaust wasted is simply that due to cylinder losses, that is, 15 per cent of the steam supplied to the engine. When the available exhaust exceeds the heating requirements for any period, the total waste is equal to

the cylinder losses plus the difference between available exhaust and the steam requirements. For example, in the first line of Table XCV., column 5 shows the steam required per hour for heating to be 5,130 pounds. The available exhaust during the day run is only 4,500 pounds; hence, the waste is simply that due to cylinder losses, or 800 pounds per hour.

During the evening run the heating requirements call for 5,400 pounds of steam per hour, and the available exhaust for the same period is 5,500 pounds. The waste from cylinder losses is 1,000 pounds; hence, the total waste is $(5,500 - 5,400) + 1,000 = 1,100$ pounds per hour. During the night the available exhaust is only 1,250 pounds per hour, which is much less than the heating requirements, so that the waste is only that due to the cylinder loss, or 225 pounds per hour.

Column 7 gives the sum of the three quantities in column 6 and represents the steam wasted in 3 hours, one from each period. Column 8 gives the steam wasted in the corresponding 30-day period, and is found by multiplying the quantity in column 6 by (8×30) .

451. *Cost of Fuel.*—The total at the bottom of column 8 is the pounds of exhaust steam wasted during the entire heating season of 210 days, and is chargeable to the fuel account for power. During the remainder of the year, 155 days, all of the exhaust is thrown away, which means that the entire quantity of steam supplied to the generator engines must be charged to the fuel account.

Assuming the power requirements for the summer season to be as follows:

	I.H.P.
Day load.....	175
Evening load.....	200
Night load.....	50

The weight of steam required per day of 24 hours will be:

$$\begin{aligned} 175 \times 30 \times 8 &= 42,000 \\ 200 \times 30 \times 8 &= 48,000 \\ 50 \times 30 \times 8 &= 12,000 \end{aligned}$$

Total.....102,000 pounds.

or $102,000 \times 155 = 15,810,000$ pounds for the summer months, making a total of $3,906,000 + 15,810,000 = 19,716,000$ pounds of

steam per year to be charged to the power plant. Assuming an evaporation of 8 pounds of steam per pound of coal, the fuel cost at \$3 per ton will be

$$\frac{19,716,000}{8 \times 2,000} \times 3 = \$3,696.75 \text{ per year.}$$

452. *Cost of Water.*—The cost of water at 5 cents per 1,000 gallons will be

$$\frac{19,716,000}{8.3 \times 1,000} \times 0.05 = \$118.77.$$

453. *Cost of Labor.*—Changing over the furnaces from hand-fired to mechanical stokers will reduce the cost of labor in the boiler room to the extent probably of one day fireman and one night fireman, amounting to a total of about \$1,600 per year. The original plant requires day and night engineers to care for the ice machine, elevator machinery, motors, etc. The new plant will probably require the services of a chief engineer in addition to these, and whose salary will be practically paid by the reduction in the boiler room force. It, therefore, seems reasonable to assume that the extra cost for labor will not exceed \$600 per year under these conditions.

454. *Total Cost of Operation.*—The total cost of operating the generating plant may be taken as follows: Depreciation being taken at 10 per cent as the plant is operated day and night to a certain extent.

Interest and depreciation on investment, 15 per cent	\$2,625.00
Taxes and insurance, 2 per cent	350.00
Repairs and supplies.	600.00
Coal.	3,696.75
Water.	118.77
Extra labor.	600.00
	\$7,990.52

455. *Cost of Power When Purchased.*—In computing the cost of generating the power upon the premises, all requirements have been reduced to indicated horse power at the main engine, upon a basis of 1.75 I. H. P. per kilowatt. Making the same assump-

tion in the present case, we find the current required per day during the *heating* season to be:

$$\begin{array}{r} 175 \\ \hline 1.75 \end{array} \times 8 = 800$$

$$\begin{array}{r} 215 \\ \hline 1.75 \end{array} \times 8 = 983$$

$$\begin{array}{r} 50 \\ \hline 1.75 \end{array} \times 8 = 229$$

2,012 kilowatt hours,

which for the entire season amounts to $2,012 \times 210 = 422,520$ kilowatt hours.

For the *summer* season the current requirements per day are as follows:

$$\begin{array}{r} 175 \\ \hline 1.75 \end{array} \times 8 = 800$$

$$\begin{array}{r} 200 \\ \hline 1.75 \end{array} \times 8 = 914$$

$$\begin{array}{r} 50 \\ \hline 1.75 \end{array} \times 8 = 229$$

1,943 kilowatt hours,

and for the season, $1,943 \times 155 = 301,165$ kilowatt hours.

The total cost per year at 2 cents per kilowatt hour is $(422,520 + 301,165) \times 0.02 = \$14,473.70$.

The saving in cost per year by the isolated plant is $\$14,473.70 - \$7,990.52 = \$6,483.18$. This large return upon the investment is due partly to the simplicity of the remodeled plant and partly to the large proportion of exhaust steam which may be utilized for heating. In this estimate no account has been taken of the saving made by the burning of a cheaper grade of coal, which is made possible by the use of mechanical stokers and forced draft. The coal used by the original plant may be estimated approximately as follows:

Refrigeration, 8,760 hours, at 714 pounds of steam per hour, 6,255,000 pounds. Laundry engine, 2,400 hours, at 800 pounds of steam per hour, 1,920,000 pounds. Mangles, 1,500 hours, at 400 pounds of steam per hour, 600,000 pounds. Heating and

ventilating 5,000 hours, at 6,000 pounds of steam per hour (average for the season), 30,000,000 pounds, making a total of $6,255,000 + 1,920,000 + 600,000 + 30,000,000 = 38,775,000$ pounds of steam per year, the evaporation of which calls for

$$\frac{38,775,000}{8 \times 2,000} = 2,383 \text{ tons of coal.}$$

If the coal burned in the old plant cost \$4 per ton, as against \$3 in the new, the saving in fuel due to the lower cost will be \$2,383, making the total saving by use of the new plant, $6,483.18 + 2,383 = \$8,866.18$ per year.

Problem (3)—Machine Shop.

456. General Proposition.—This problem deals with a large machine shop having a steam power plant of only sufficient capacity for driving the machinery. At present the buildings are lighted with gas, but it has been decided to change over to electricity. Will it be better to purchase current from a local company at 7 cents per kilowatt hour or install a private plant and generate it upon the premises?

There are four buildings, 50x150 feet in size, and three stories in height, containing a total of 10,200 square feet of direct steam coils. The power plant consists of a 150-H. P. high-speed four-valve simple engine, using 28 pounds of steam per I. H. P. per hour, and three 75-H. P. boilers of the return tubular type, equipped with mechanical stokers and forced draft. The exhaust steam is utilized for heating purposes. There is a river adjacent to the plant, so that water can be had for condensing purposes for the cost of pumping, but is not made use of at present. Power is transmitted to the different buildings electrically, and the line shafting is motor-driven.

457. Steam Required for Heating.—The first step is to determine if any portion of the exhaust from the new generator engine can be utilized for heating purposes. This information could probably be learned at the plant, but in any case can be computed with sufficient accuracy as follows:

Radiation of the type used will condense approximately one-third of a pound of steam per square foot of surface per hour under average conditions, which also includes the heat loss from

the supply and return mains. This gives a maximum steam consumption for heating of $10,200 \div 3 = 3,400$ pounds per hour for the entire plant.

The available exhaust from the present engine is $150 \times 28 \times 0.85 = 3,570$ pounds per hour, which is evidently ample for all heating purposes in the coldest weather, and shows that the exhaust from the new generating engine must be wasted.

458. Lighting Plant.—As the building must be wired in either case, whether the current is purchased or generated, this item drops out of the problem although lamps must be furnished in case of the private plant. Each building has a floor space of $50 \times 150 \times 3 = 22,500$ square feet, making a total of $22,500 \times 4 = 90,000$ square feet for the entire plant. Assuming that closed arc lamps with opal globes are to be generally used, we find from Table XLVII., Paragraph 193, that 0.75 watts will be required per square foot of floor space lighted, making a total of

$$\frac{90,000 \times 0.75}{1,000} = 67.5 \text{ kilowatts.}$$

In addition to the above there will be a considerable number of incandescent lamps for offices and for the individual lighting of special machines which require particularly good illumination, making it necessary to install an 80-kilowatt machine driven by an engine of $80 \times 1.75 = 140$ I. H. P. capacity. As the buildings are equipped for gas, and the plant will not be operated at full load for more than about $2\frac{1}{2}$ hours at one time, a single unit may be installed for the regular work; this to be supplemented by a 25-kilowatt outfit for use on dark days and in the summer season when light is only required at special machines for a short time morning and night.

As the exhaust cannot be utilized, and as the larger machine will run at practically full load, it would seem best to first investigate the matter of steam economy, with a view to the possible installation of a compound condensing engine. The conditions at first sight appear to be especially favorable for a condensing outfit because it could be used with the lighting engine during the heating season, and with the power engine through the summer when the larger lighting unit was not in use.

The smaller engine may be of the simple high-speed type, piped to run either condensing or non-condensing, as desired.

459. *Boiler Plant.*—There are three ways in which the capacity of the boiler plant may be increased, as follows: Adding new boilers; increasing the rate of combustion under the present boilers, and by installing an economizer. It would seem after a careful examination of the problem, that a combination of the three might work to advantage. For example: Suppose the present boilers are working under a draft which produces a combustion of 15 pounds of coal per hour per square foot of grate. This might easily be raised to 18 or 20 pounds without any appreciable loss in efficiency, especially if an economizer were installed.

The present boiler plant, with its reserve unit, has a rating of $3 \times 75 = 225$ H. P. Increasing the combustion 20 per cent, will raise the capacity to $225 \times 1.2 = 270$ H. P.

The effect of an economizer, while burning the same amount of fuel, should be to increase the boiler capacity at least 10 per cent, making a total of $270 \times 1.1 = 297$ H. P. If a compound condensing medium-speed engine is used for lighting, with a steam consumption of 17 pounds per I. H. P. per hour, the total steam required will be $(140 \times 17) + (150 \times 28) = 6,580$ pounds per hour, which calls for a boiler capacity of approximately $6,580 \div 30 = 220$ H. P. The auxiliaries will require about 20 H. P. leaving a margin of $297 - (220 + 20) = 57$ H. P. While this will provide ample capacity without the addition of another boiler it is best in a plant of this size to install a duplicate unit to be held in reserve in case of emergency.

460. *Cost of Lighting Plant.*—The cost of additional equipment for the lighting plant, exclusive of switchboard, wiring and lamps, may be taken as follows:

Economizer	\$1,500
Boiler and stoker	975
Compound medium-speed engine	2,800
Simple high-speed engine	600
Dynamo (80 Kw.)	1,450
Dynamo (25 Kw.)	375
Condensing apparatus and pipe line to river	1,500
Steam piping, etc.	2,000
Foundations	800
Total	\$12,000

461. *Cost of Operation.*—In estimating the number of hours which the entire lighting plant will be in operation on clear days we may assume the following, the working hours extending from 7 a. m. until 6 p. m.;

Month.	Lighting hours.		Total.
	Morning.	Night.	
September.....	..	1	1
October.....	..	1½	1½
November.....	½	2	2½
December.....	1	2	3
January.....	1	1½	2½
February.....	½	1	1½
March.....	½	½	1

Omitting Sundays this amounts to 335 hours per year, which, however, should be increased about one-third for cloudy weather and for day lighting in special parts of the plant. This gives $335 \times 1.33 = 446$, which, in round numbers, may be taken as 450 hours, calling for $140 \times 450 = 63,000$ horse-power hours, or

$$\frac{63,000}{1.75} = 36,000 \text{ kilowatt hours.}$$

Assuming 80 per cent of the work to be done by the larger engine, on a basis of 17 pounds of steam per I. H. P. per hour, and 20 per cent by the smaller engine on a basis of 26 pounds per I. H. P., calls for $(63,000 \times 0.80 \times 17) + (63,000 \times 0.20 \times 26) = 1,184,400$ pounds of steam per year for lighting. This with an evaporation of 9 pounds of steam per pound of coal (which should be obtained with the economizer), and coal at \$3 per ton will cost

$$\frac{1,184,400}{9 \times 2,000} \times 3 = \$197.40.$$

With a jet condenser the cooling water, at 30 pounds per pound of steam, will amount to $1,184,400 \times 30 = 35,532,000$ pounds, or $35,532,000 \div 8.3 = 4,300,000$ gallons, in round numbers. The cost of pumping the water from the river may be taken as \$0.008 per 1,000 gallons, which amounts to $4,300 \times 0.008 = \$34.40$. As the plant is equipped with mechanical stokers and forced draft, no additional help will be required in the boiler room on account of the extra boiler for the short time it will be in use night and morning. One first-class engineer can attend to the additional engine, with a small amount of help for cleaning, etc., and we will

assume that this is available without extra cost. A man will be necessary in any case to have general care of the lamps and he can also furnish whatever assistance is necessary in the engine room. On this basis we have the following items making up the operating cost of the lighting plant, from boilers to switch-board. Depreciation is taken as only 4 per cent, as the plant is operated for only a portion of the time.

Interest and depreciation, on investment 9%,	\$1,080.00
Taxes and insurance, 2%.....	240.00
Repairs and supplies.....	150.00
Water for condensing.....	34.40
Coal.....	197.40
Total.....	\$1,701.80

462. Saving in Fuel for Power.—The main shop engine runs 3,000 hours per year, and under the original plan requires $150 \times 28 \times 3,000 = 12,600,000$ pounds of steam. Assuming an evaporation of 8 pounds of steam per pound of coal this calls for

$$\frac{12,600,000}{8 \times 2,000} = 788 \text{ tons per year.}$$

It has been assumed that the addition of an economizer will increase the power of the boiler plant 10 per cent, while burning a given amount of coal, which also means a saving of 10 per cent in the fuel burned for generating a given amount of steam. As the entire cost of operating the economizer has been charged to the lighting account, we may deduct 10 per cent from the coal consumption of the shop plant which profits by the use of the economizer throughout the year. This reduces the coal required for power purposes $788 \times 0.10 = 79$ tons, in round numbers. Furthermore, the condensing apparatus may be used in connection with the shop engine for at least 180 days per year, reducing the steam consumption from 28 pounds per I. H. P. per hour to about 24, and saving $150 \times 4 \times 10 \times 180 = 1,080,000$ pounds, or its equivalent in coal of

$$\frac{1,080,000}{9 \times 2,000} = 60 \text{ tons,}$$

thus reducing the yearly supply for shop power $79 + 60 = 139$ tons. This produces a saving of $139 \times 3 = \$417$ per year. From this must be deducted the cost of pumping the condensing water,

which, for 180 days amounts to $(150 \times 24 \times 30 \times 10 \times 180) \div 8.3 = 23,421,700$ gallons, which at \$0.008 per 1,000 gallons, costs $23,421.7 \times 0.008 = \$187.37$, leaving $417 - 187.37 = \$229.63$ as the saving in fuel. This deducted from the operating cost of the lighting plant leaves a net cost of $1,701.80 - 229.63 = \$1,472.17$ per year for lighting.

The cost of purchased current would be $36,000 \times 0.07 = \$2,520$, leaving a balance of $2,520.00 - 1,472.17 = \$1,047.83$ in favor of the private plant, which pays

$$\frac{1,047.83}{12,000} = 0.0873 \text{ or } 8.73 \text{ per cent}$$

upon the cost of installation.

This plant operates under two decided disadvantages and still shows a good return upon the investment. No part of the exhaust can be utilized for heating, and again, the total output per year is small compared with the cost of the plant, the lighting engine running only 450 hours per year as against 3,000 hours in case of the main power or shop engine.

463. Alternative Scheme.—Let us now see what results might be expected by omitting the economizer and condenser and installing simple high-speed engines using 30 pounds of steam per I. H. P. per hour and exhausting outboard. In this case one new boiler will be provided as before. While this does not leave an extra unit for emergencies, the plant is equipped with forced draft and three of the boilers may be made to do the entire work temporarily by forcing them about 30 per cent, which is entirely practicable. This, with gas lighting to fall back upon, seems to be amply safe for ordinary conditions.

464. Cost of Equipment.—The cost of this arrangement will be approximately as follows:

Boiler and stoker.....	\$975
Engine, 140 H.P.....	2,000
Engine, 40 H.P.....	600
Dynamo, 80 Kw.....	1,450
Dynamo, 25 Kw.....	375
Piping.....	1,800
Foundations.....	800
Total.....	<u>\$8,000</u>

465. Cost of Fuel.—The cost of fuel in this case will be as follows: $140 \times 450 \times 30 = 1,890,000$ pounds of steam per year,

which, with an evaporation of 8 pounds per pound of coal, and coal at \$3 per ton, will amount to

$$\frac{1,890,000}{8 \times 2,000} \times 3 = \$354.38.$$

466. *Cost of Operation.*—The cost of operation will therefore be:

Interest and depreciation on investment, 9%	\$720.00
Taxes and insurance, 2%.....	160.00
Repairs and supplies.....	150.00
Coal.....	354.38
Total.....	<u>\$1,384.38</u>

467. *Saving in Cost of Current.*—The saving over purchased current in this case is $2,520 - 1,384.38 = \$1,135.62$, which pays

$$\frac{1,135.62}{8,000} = 0.142 \text{ or } 14.2 \text{ per cent}$$

on the investment.

As no change has been made in the main or shop power plant, that does not enter into the computation.

468. *Necessity of Investigation.*—This problem is of special interest as showing the necessity of investigating a proposition from all sides in order to determine the best method of procedure.

While a superficial examination seems to favor the use of an economizer and condensing outfit in the above problem, a more careful study shows a simple and less expensive equipment to be the more economical from an operating standpoint, even at a considerable increase in the cost of fuel.

469. *Lay-Out of Plant.*—Figs. 217 and 218 show in diagram the original and remodeled plants respectively, for the proposition under consideration, it being decided to use the cheaper outfit. This arrangement works out in a very simple manner, especially when there is space for the new apparatus in the boiler and engine rooms, as has been assumed in the present case. No changes have been made in the old plant except to lengthen the main steam drum over the boilers and connect the new unit with the same. A separate line of steam piping is run to the new engines and the exhaust is carried outboard independently. No change is made in the feed-water heater, but a branch is carried from the new exhaust pipe to the old, in order to furnish steam

for feed-water heating in the very coldest weather, when practically all of the exhaust from the shop engine is utilized for heating purposes. This condition will seldom occur, but the expense of making the connection is small and it is well to provide it for possible use.

As the power required for driving the machinery and for lighting is so equally divided, it might be a better plan to duplicate the present engine and generator, thus making the units interchangeable. In this case either could be used during the day and the

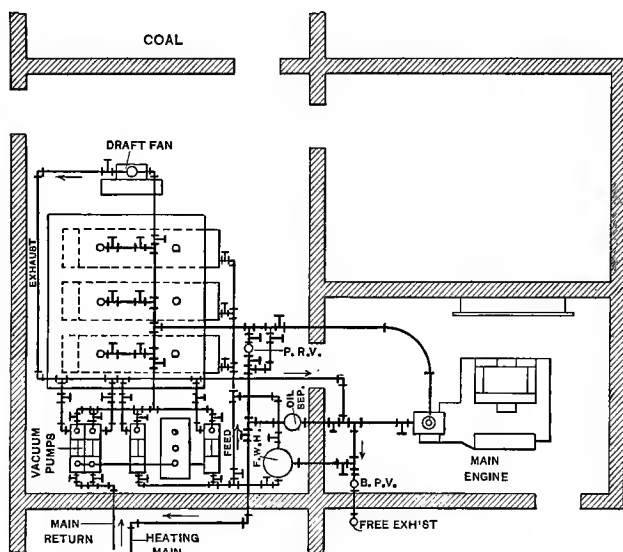


Fig. 217. Machine Shop—Original Plant.

second started night and morning when lights were required. It might also be more desirable in this case to enlarge the supply and exhaust piping than to run duplicate lines, although somewhat more expensive.

Problem (4)—Hospital.

470. *General Proposition.*—The problem considered in this case relates to a hospital plant made up of detached buildings, as shown in Fig. 219.

The institution has been built up gradually, one or two buildings at a time, without reference to a central power and heating plant. At present steam for heating purposes is generated in five separate boilers, or batteries of boilers, of the cast-iron sectional type, using anthracite coal at \$6 per ton. Buildings Nos. 1, 2 and 3 each have an individual heating plant, while buildings Nos. 4, 6, 8 and 5, 7, 9 are supplied by two batteries of boilers located in buildings 4 and 5, respectively. Building 1 also has a

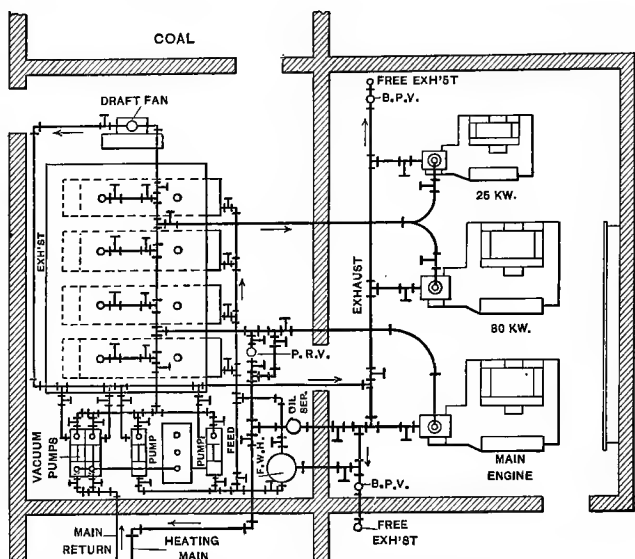


Fig. 218. Machine Shop—Remodeled Plant.

small high-pressure boiler furnishing steam for laundry purposes, cooking, sterilizing, etc. Current for lighting, refrigeration, and for operating a small elevator, fans, kitchen motor, etc., is purchased from a local company at a cost of 3 cents per kilowatt hour.

It is now proposed to erect a power house at the point shown, the upper floor of which shall serve as a laundry, and install a power and lighting plant for supplying all of the buildings. The problem involves the determination of the sizes of the various

parts of the apparatus comprising the proposed plant, its approximate cost and the expense of operation as compared with the present arrangement.

471. *Increased Economy.*—Economy of operation may be expected in three ways, as follows: First, a higher boiler efficiency, due to larger units and more careful firing; second, a cheaper grade of coal may be used with properly designed fur-

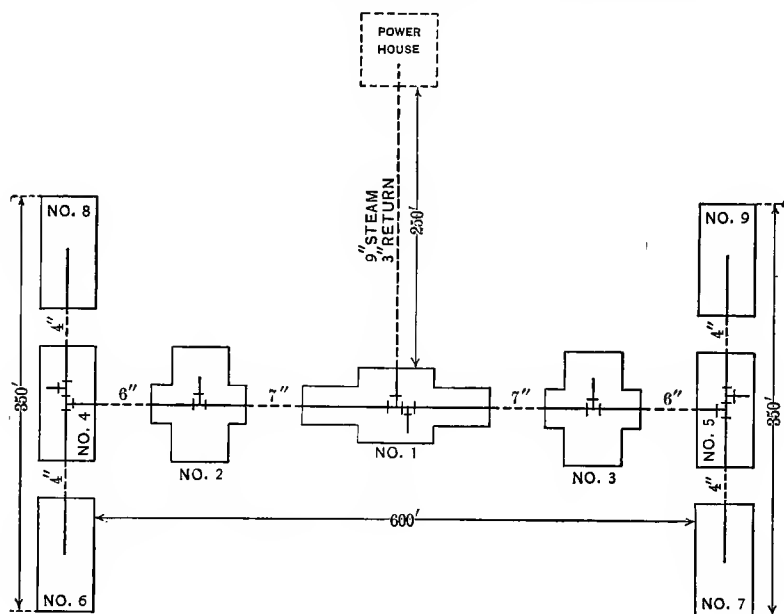


Fig. 219. Block Plan of Hospital Buildings.

naces and the stronger draft which will be available; third, the exhaust steam, or a considerable portion of it, may be used for heating purposes during the winter, thus reducing the cost of electricity to a point somewhat below that paid the local company. In addition to these are several minor advantages, such as a reduction in the rate of insurance due to the removal of the furnaces from the different buildings; the installation of a vacuum system of steam circulation, resulting in more efficient temperature regulation; the avoidance of dust and noise in the ward buildings; and

making available a certain amount of space previously occupied by boilers which may now be devoted to other useful purposes.

472. *Boiler Capacity for Heating.*—The radiating surface in the different buildings is as follows:

TABLE XCVI.

No. of building.	. Square feet of	
	Direct surface.	Indirect surface.
1.....	700	2,030
2.....	370	1,560
3.....	370	1,560
4.....	280	900
5.....	280	900
6.....	280	900
7.....	280	900
8.....	280	900
9.....	280	900
Total.....	3,120	10,550

Assuming efficiencies of 250 and 400 B. T. U. per square foot of surface per hour, in zero weather, for direct and indirect radiation respectively, we have a total of $(3,120 \times 250) + (10,550 \times 400) = 5,000,000$ B. T. U. which calls for $5,000,000 \div 30,000 = 166.6$ boiler horse power. Increasing this 20 per cent for losses from pipe mains, etc., gives $166.6 \times 1.2 = 200$ H. P. Assuming a maximum of 30 H. P. for laundry work, hot-water service, etc., makes a total of $200 + 30 = 230$ boiler horse power required for all heating purposes in the coldest weather. Although 30 H. P. must be provided for laundry purposes, in order to cover maximum requirements for comparatively short periods, a *continuous* output of 20 H. P. is all that need be provided for when computing the steam consumption, owing to the storage of heat during the night in a large hot-water tank.

473. *Boiler Capacity for Power.*—The refrigerating machine is motor-driven and rated at 5.5 tons per 24 hours. This, from Paragraph 195, calls for an I. H. P. at the generator engine of $5.5 \times 1.8 = 10$. The various other motors are required to deliver 28 H. P., which amounts to $28 \div 0.7 = 40$ I. H. P. at the main engine after allowing for the efficiency losses in the various machines. (See Paragraph 192.)

The maximum lighting load, including the grounds, is 60 kilowatts, calling for $60 \times 1.75 = 105$ I. H. P. at the generator en-

gine. This makes a total of $10+40+105 = 155$ I. H. P., which for a simple high-speed engine will require $155 \times 30 = 4,650$ pounds of steam per hour, or $4,650 \div 30 = 155$ boiler horse power.

As this is less than the boiler power required for heating, all of the exhaust steam can be utilized in the coldest weather, so that only the cylinder losses need be considered in connection with the power work when computing the total boiler capacity. Assuming these losses to be 15 per cent of the steam supplied to the engine, $155 \times 0.15 = 23$ H. P. must be provided in addition to that already computed for heating purposes, making a total of $230+23 = 253$ H. P. to be installed in the new power house.

It will be best in a plant of this size to provide a spare boiler, hence, in making the lay-out and estimating the cost we will arrange for five 64-H. P. boilers.

474. *Engines and Generators.*—In computing the daily load the 24 hours may be divided into three periods, as follows, during the heating season:

DAY LOAD (10 HRS.)	
	I.H.P.
Refrigeration.....	10
Motors.....	40
Lighting (10 Kw.).....	17
. Total.....	67

EVENING LOAD (5 HRS.)	
	I.H.P.
Refrigeration.....	10
Motors.....	10
Lighting (60 Kw.).....	105
Total.....	125

NIGHT LOAD (9 HRS.)	
	I.H.P.
Refrigeration.....	10
Lighting (25 Kw.).....	44
Total.....	54

If two 40-kilowatt generators with 70-H. P. direct-connected engines are provided, one will be sufficient for the day and night loads, and both running together will care for the evening load; the apparatus in each case operating within economical limits. As a matter of precaution it will be well to provide a third unit of the same size as a reserve, in case of accident.

475. *Cost of New Plant.*—In estimating the cost, a portion of the new power house should be charged to the general building fund as the upper floor is to be used as a laundry. The pipe mains from the boiler house to building No. 1 are to be carried in a conduit of new construction, but the remainder of the way will run through basements and open corridors. The sizes of the supply

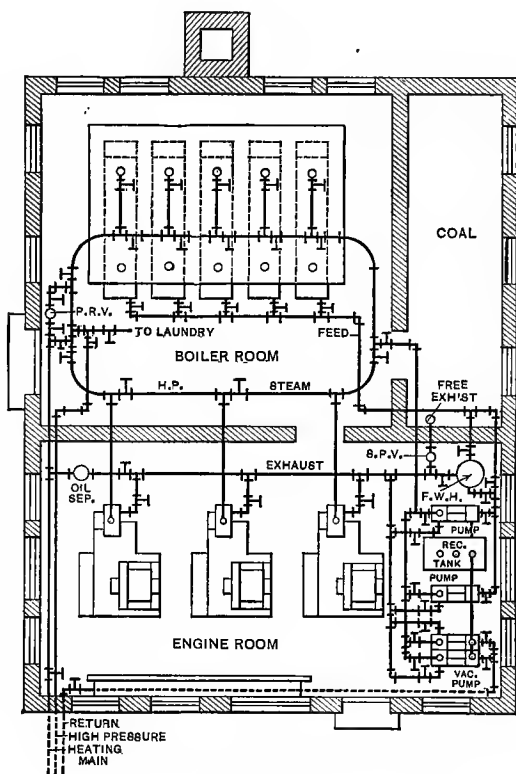


Fig. 220. Power and Heating Plant of Hospital.

mains are indicated in Fig. 219. The returns are not shown, but follow the general line of the steam mains. The present boilers may be removed, if the additional space is required, and valved connections made between the new mains and the present supply and return piping in each building. Special vacuum controllers

(see Fig. 69), should be placed in the return from each group of buildings in order to equalize the suction throughout the system. A general lay-out of the new plant is shown in diagram in Fig. 220. The approximate cost may be estimated as follows:

Share of building (two-thirds)	\$5,000
Chimney and smoke-flue	1,000
Pipe lines and conduit	2,200
Boilers (5)	3,500
Engines (3)	3,000
Generators (3)	1,800
Switchboard	800
Boiler room piping, pumps and heaters	3,500
Foundations	1,200
Total	\$22,000

476. *Power and Heating Requirements.*—In computing the cost of operation for this locality, the year may be divided into two parts, a heating season of 270 days and a non-heating season of 95 days. We will divide the heating season into 9 periods of 30 days each and determine the average temperature of each period from the weather records of previous years. Assume these to be as given in Table XCVII., column 2.

TABLE XCVII.

1. Approximate period of the year.	2. Average temperature, degrees.	3. Factor for temper- ature regula- tion.	4. Boiler H.P. for heating and hot-water service.
1. September	60	0.31	82
2. October	50	0.43	106
3. November	40	0.54	128
4. December	30	0.66	152
5. January	20	0.77	174
6. February	30	0.66	152
7. March	40	0.54	128
8. April	50	0.43	106
9. May	60	0.31	82

It will be noticed that the factors in column 3 have been taken from curve No. 3, Fig. 211, which is for a hot-water system. Although the plant is to be heated by steam under a vacuum circulation, a considerable portion of the radiating surface is of the indirect type provided with mixing dampers, so that a much closer temperature regulation is possible than with the usual direct system, and the conditions will correspond more closely with the

factors given by curve No. 3 than by curve No. 4. Furthermore, results relating to steam quantities, are, for simplicity, carried through in terms of boiler horse power instead of pounds of steam, the term "horse power" in this connection meaning the equivalent of 30 pounds of steam per hour. Also, with the type of engine used, the steam furnished by one boiler horse power and that required by one engine horse power are identical (30 pounds per hour).

It has already been shown that the day load for 10 hours amounts to 67 I. H. P., which means that the equivalent of $67 \times 0.15 = 10$ boiler horse power are consumed in cylinder losses and that $67 \times 0.85 = 57$ H. P. are available for heating in the form of exhaust steam. Table XCVII., column 4, gives the boiler horse power required for heating and hot-water service for the different periods. These quantities are equal to 200 multiplied by the temperature factor in column 3 plus 20, and show that the heating requirements during the day exceed the available exhaust in each period during the heating season. Table XCVIII. shows the boiler horse power required for both heating and power during the day, evening and night, for each of the nine periods of the heating season. For example, during the day, in period 1, the requirements for heating amount to 82 H. P. (see Table XCVII.), 57 H. P. of

TABLE XCVIII.
FOR HEATING SEASON.

Period	Day, 10 hours.		Evening, 5 hours.		Night, 9 hours.		30-day period.	
	H.P. for heating.	H.P. for power.	H.P. for heating.	H.P. for power.	H.P. for heating.	H.P. for power.	H.P. hours for heating.	H.P. hours for power.
1.	82	10	82	43	82	8	59,040	11,610
2.	106	10	106	19	106	8	76,320	8,010
3.	128	10	128	19	128	8	92,160	8,010
4.	152	10	152	19	152	8	109,440	8,010
5.	174	10	174	19	174	8	125,280	8,010
6.	152	10	152	19	152	8	109,440	8,010
7.	128	10	128	19	128	8	92,160	8,010
8.	106	10	106	19	106	8	76,320	8,010
9.	82	10	82	43	82	8	59,040	11,610
Total H.P. hours for the heating season.							799,200	79,290

this is available in the exhaust, and $82 - 57 = 25$ H. P. must be made up by live steam. The total steam necessary for both heating and power is $82 + 10 = 92$ H. P. Of this, 82 H. P. should

be charged to the heating account and 10 H. P. (the cylinder losses) to the power account. During the evenings of period 1 the power load amounts to 125 I. H. P., with cylinder losses of $125 \times 0.15 = 19$ H. P. and available exhaust equivalent to $125 \times 0.85 = 106$ H. P. In this case $106 - 82 = 24$ H. P. are exhausted outboard, and the total to be charged to power is $19 + 24 = 43$ H. P., while that charged to heating remains the same as before. During the night the power load is 54 I. H. P. with cylinder losses of $54 \times 0.15 = 8$ H. P. and available exhaust equivalent to $54 \times 0.85 = 46$ H. P. Here the exhaust is less than the heating requirements and 8 H. P. will be chargeable to power and 82 H. P. to heating.

In like manner the same process is carried out for an average day in each period during the season.

The figures in the last two columns are the horse-power hours, for heat and power, respectively, for each 30-day period. For example, $82 \times 24 \times 30 = 59,040$ -H. P. hours for heating in the first period and $[(10 \times 10) + (5 \times 43) + (9 \times 8)] \times 30 = 11,610$ H. P. hours chargeable to power.

During the non-heating season of 95 days, 20 H. P. for hot-water service is chargeable to heating, amounting to $20 \times 24 \times 95 = 45,600$ H. P. hours.

The horse power chargeable to the power account is that required by the engines, less 20 H. P., which is the exhaust utilized for heating, as noted above. Here, also, the days are longer than during the winter and the full number of lights will not be required for so many hours. We may provide for this by reducing the time of the evening load to 3 hours and increasing the night load to 11 hours.

We have, therefore, for a single day, the following:

	H.P. hours.
Day load (67 — 20) $\times$ 10 = ...	470
Evening load (125 — 20) $\times$ 3 = ...	315
Night load (54 — 20) $\times$ 11 = ...	374
Total	1,159

or $1,159 \times 95 = 110,105$ H. P. hours for the non-heating season.

477. *Cost of Operation.*—The fuel cost for heating may be estimated on a basis of 4 pounds of coal per horse power per hour, which should easily be obtained with a plant of this kind. This,

with coal at \$4 per ton, the price paid for the grade used in the new plant will amount to

$$\frac{(799,200 + 45,600) \times 4}{2,000} \times 4 = \$6,758.$$

Fuel cost for *power*,

$$\frac{(79,290 + 110,105) \times 4}{2,000} \times 4 = \$1,515.$$

The new plant will require a chief engineer at a salary of, say \$1,600 per year, and an assistant for night work at about \$1,200. On the other hand, less help will be required for the single larger boiler plant than for the five smaller ones, and it seems safe to say that one man may be dispensed with, when the handling of both coal and ashes is considered. This will result in a saving of about \$700, leaving an additional cost for labor of $(1,600 + 1,200) - 700 = \$2,100$.

The cost, therefore, of operating the combined power and heating plant for a year will be approximately as follows:

Interest and depreciation on investment, 15%	\$3,300
Taxes and insurance, 1.5%	330
Additional labor	2,100
Repairs and supplies	400
Coal (\$6,758 + \$1,515)	8,273
Total	\$14,403

In the old plant with scattered boilers, and less efficient handling, probably about 5 pounds of coal would be required per horse power per hour, making the fuel cost for the year, with coal at \$6 per ton,

$$\frac{(799,200 + 45,600) \times 5}{2,000} \times 6 = \$12,672$$

The cost of purchased current may be computed as follows:

HEATING SEASON.			H.P. hrs.
Day load	67	$\times 10 =$	670
Evening load	125	$\times 5 =$	625
Night load	54	$\times 9 =$	486
Total for 24 hours			1,781

NON-HEATING SEASON

Day load	67	×	10	=	670
Evening load.....	125	×	3	=	375
Night load.....	54	×	11	=	594
					1,639
Total for 24 hours					1,639

Total for the year $(1,781 \times 270) + (1,639 \times 95) = 636,575$
H. P. hours.

As the horse-power ratings used in the problem have been, for the most part, based upon 1.75 I. H. P. per kilowatt, we may use the same ratio in changing horse-power hours into kilowatt hours with sufficient accuracy. This gives a total of $636,575 \div 1.75 = 363,757$ -kilowatt hours, which at 3 cents per kilowatt hour will cost $363,757 \times 0.03 = \$10,913$, making the total for both power and heating $10,913 + 12,672 = \$23,585$ per year.

The saving, therefore, by the new plant is $23,585 - 14,403 = \$9,182$. This large saving is due to the continuous use of a large part of the plant, the utilization of all of the exhaust steam, the greater efficiency of the boiler plant and the cheaper grade of coal used.

If it were desired to carry through the last computation in kilowatt hours instead of horse-power hours the method would be practically the same.

In this case the power required for refrigeration would be $1.2 \times 5.5 \times 0.746 = 4.9$ kilowatts, and for the small motors, $\frac{28 \times 0.746}{0.85}$

$= 24.6$ kilowatts for the day load, and $\frac{24.6}{4} = 6.1$ kilowatts for the evening load.

The lighting load for each period is given in kilowatts in the problem.

Makes of Equipment, or Catalogues from Which the Following Cuts Were Taken:

- | | |
|--|--|
| <p>Fig. 3 American Radiator Co.
 " 15 American Radiator Co.
 " 22 American Radiator Co.
 " 92 Central Station Steam Co.
 " 95 Johns-Manville Co.
 " 96 Percy N. Kenway.
 " 97 Michigan Pipe Co.
 " 98 A. Wyckoff & Son Co.
 " 103 Heine Boiler Co.
 " 104 Wilkinson Stoker.
 " 106 Taylor Stoker.
 " 107 Heine Boiler Co.
 " 108 Green Fuel Economizer Co.
 " 114 Link Belt Co.
 " 115 C. W. Hunt Co.
 " 116 C. W. Hunt Co.
 " 120 Green Fuel Economizer Co.
 " 121 Green Fuel Economizer Co.
 " 122 Ames Iron Works.
 " 123 Ames Iron Works.
 " 124 Ball Engine Co.
 " 125 American Ball Engine Co.
 " 126 Sturtevant Turbine.
 " 127 Terry Turbine.
 " 128 Kerr Turbine.
 " 129 Rateau Turbine.
 " 130 Westinghouse-Parsons Turbine.
 " 131 Westinghouse Turbine.
 " 135 Davidson Pump.
 " 137 Blake-Knowles Pump.</p> | <p>Fig. 138 Deane Pump.
 " 139 Worthington Pump.
 " 140 Worthington Pump.
 " 141 Blake-Knowles Pump.
 " 142 Deming Pump.
 " 143 Deming Pump.
 " 144 Blake-Knowles Pump.
 " 149 Blake-Knowles Pump.
 " 150 Blake-Knowles Pump.
 " 151 Blake-Knowles Pump.
 " 152 Janesville Iron Works.
 " 153 Cochrane Heater.
 " 154 Cochrane Heater.
 " 157 Worthington Condenser.
 " 158 Blake-Knowles Condenser.
 " 159 Worthington Condenser.
 " 160 Westinghouse-Leblanc Condenser.
 " 161 Westinghouse-Leblanc Condenser.
 " 162 Westinghouse-Leblanc Pump.
 " 163 Blake-Knowles Condenser.
 " 164 Blake-Knowles Condenser.
 " 165 Worthington Condenser.
 " 166 Wheeler Condenser.
 " 167 Wheeler Condenser.
 " 168 Baragrath Condenser.
 " 184 Alberger Condenser Co.
 " 185 Alberger Condenser Co.
 " 186 Alberger Condenser Co.
 " 187 Alberger Condenser Co.
 " 188 Alberger Condenser Co.</p> |
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